IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

First Named Inventor	: Joseph H. Sassine	
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Filed	: February 27, 2004	Group Art Unit: 2627
Title	: HEAD SUSPENSION ASSEMBLY HAVING A HIGH DAMPING HIGH STIFFNESS COMPONENT	Examiner: Watko, Julie Ann
Docket No.	: STL11400.00_ I69.12-0600	

APPEAL BRIEF FOR APPELLANTS

Mail Stop Appeal Brief – Patents
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INTRODUCTION

This Brief is filed in support of Appellants' Notice of Appeal filed February 17, 2010, appealing from the final Office Action mailed January 27, 2010.

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Real Party in Interest

The real party in interest in this Appeal is Seagate Technology LLC of Scotts Valley, California, the assignee of record and owner of the entire right, title, and interest in the application.

Related Appeals and Interferences

There are no prior or pending appeals, judicial proceedings or interferences known to Appellants that are related to, will directly affect or be directly affected by, or have a bearing on the Board's decision in this Appeal.

Status of Claims

I. Claims

A. Claims in the application: 1–3, 5–12, 14–16, 18–20, 26–32

B. Total number of claims in the application: 24

II. Status of Claims

A. Claims canceled: 4, 13, 17, 21–25

B. Claims allowed: –None–

C. Claims rejected: 1–3, 5–12, 14–16, 18–20, 26–32

D. Claims withdrawn: –None–

E. Claims objected to: –None–

F. Pending claims: 1–3, 5–12, 14–16, 18–20, 26–32

III. Claims on Appeal

A. Claims on appeal: 1–3, 5–12, 14–16, 18–20, 26–32

B. Total claims on appeal: 24

Status of Amendments

No amendments have been made to the claims following the final Office Action mailed January 27, 2010, in which claims 1–3, 5–12, 14–16, 18–20 and 26–32 were rejected. The status of the claims is as amended December 3, 2009, following the non-final Office Action mailed September 3, 2009.

Summary of Claimed Subject Matter

The present invention relates to a head suspension assembly for a disc drive, where the suspension assembly uses a damping material to reduce high frequency vibration. Application, U.S. Pub. No. 2005/0190502, ¶ 2. An exemplary disc drive 10 is shown in FIG. 1 (below), with voice coil motor 12 to rotate actuator arm 16 about spindle axis 14. Suspension assembly 18 connects to actuator arm 16 at mounting block 20. Gimbal assembly 22 connects to the distal end of suspension 18 and carries slider 24 with a transducing head for reading and writing data on concentric tracks 29 of disc 27. *Id.*, ¶ 16.

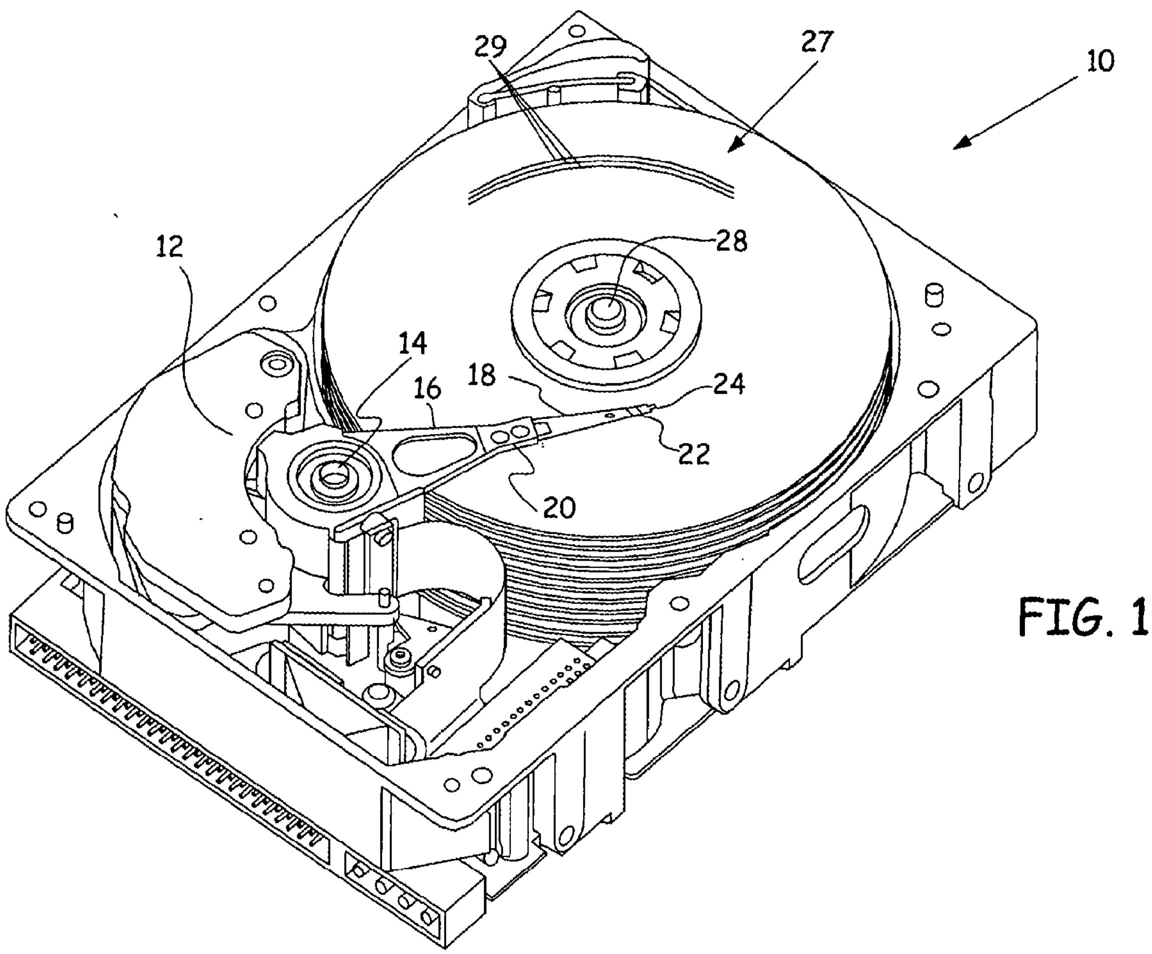

FIG. 1 of the Application

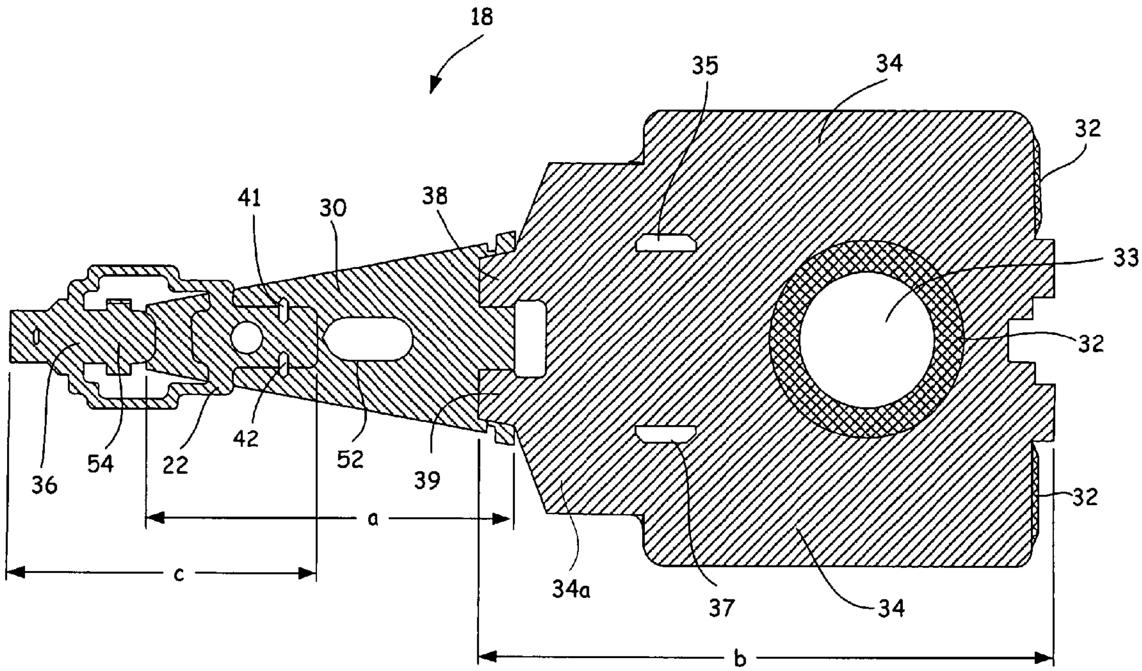

FIG. 2 of the Application

An exemplary suspension assembly 18 is shown in FIG. 2 (above), with beam component 30 spanning longitudinal range “a,” hinge component 34 spanning range “b” and gimbal component 36 spanning range “c.” *Id.*, ¶ 17. Hinge 34 attaches to the rear of beam 30 at points 38, 39, and stacks on baseplate 32 with connecting points 35, 37. Baseplate 32 has mounting hole 33 for attaching assembly 18 to actuator arm 16 (FIG. 1). Gimbal 36 attaches to the front of beam 30 at points 41, 42 and connects to slider assembly 24 (FIG. 1). *Id.*, ¶¶ 17–19.

Independent claim 1 recites a head suspension assembly [18] comprising a beam component [30] having a front end and a rear end, a hinge component [34] near the rear end of the beam component [30] for connecting to an actuation arm [16], and a gimbal component [36] near the front end of the beam component [30] for carrying a transducing head. At least one of the hinge [34] and gimbal [36] is separately made and attached to the beam [30]. *Id.*, ¶¶ 30, 32.

The hinge [34] comprises a first structural damping material and the gimbal [36] comprises a second structural damping material. *Id.*, ¶¶ 5, 8–10, 20–24, 25, 27, 28. The structural damping materials have a modulus of elasticity greater than approximately 10 gigapascals (GPa) and a damping capacity (ζ, “zeta”) greater than approximately 0.02 in a vibration mode having a

frequency between about 6,010 Hz and about 22,650 Hz. *Id.*, ¶¶ 34, 40. Damping capacity ζ is related to the logarithmic decrement (δ, "delta") by $\zeta = \delta/2\pi$. *Id.*, ¶¶ 34–36; see also ¶¶ 4, 22, 26, 27.

TABLE 1

	Units	Baseline Design	High Damping Design
First bending	Hz	6237	6010
First torsion	Hz	9027	9976
Second bending	Hz	14534	12150
Second torsion	Hz	22959	22650
Sway	Hz	19495	18645
Gimbal First bending	Hz	13527	7886
Gimbal first torsion	Hz	14050	8490
Spring rate	gram-force/in	52	56
Pitch stiffness	μNm/deg	0.75	0.4
Role stiffness	μNm/deg	0.83	0.38

Table 1 of the Application

Vibration modes of interest are disclosed in Table 1 and FIGS. 3 and 4. Table 1 (above) compares finite-element (FE) simulation results for the invention (high damping design) and a baseline design at mode frequencies from about 6,010 Hz to about 22,650 Hz. *Id.*, ¶¶ 48–51.

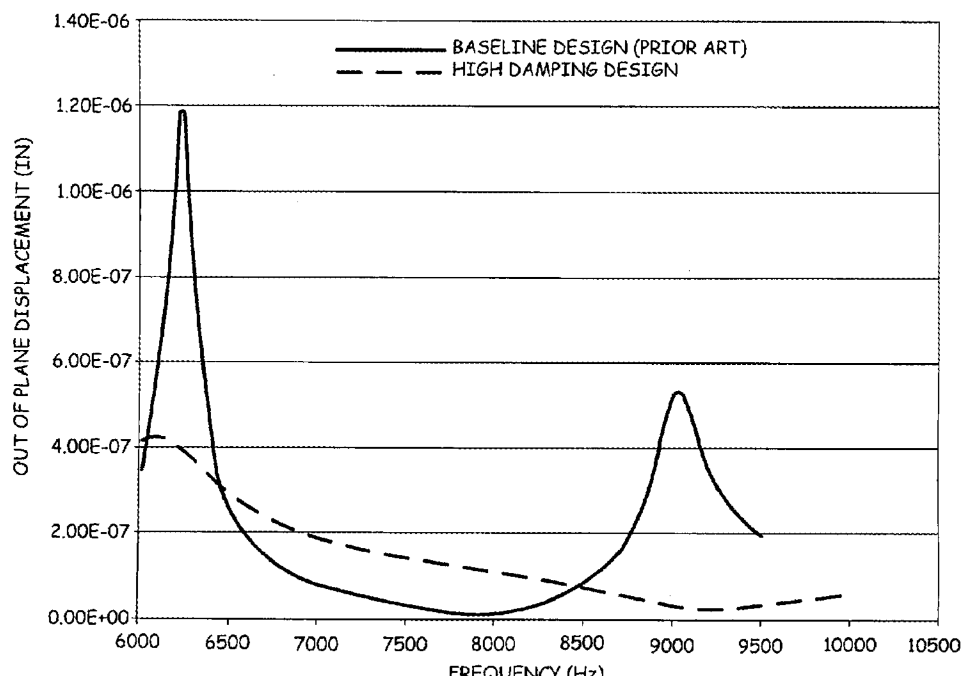

FIG. 3

FIG. 3 of the Application

FIG. 3 (above) and FIG. 4 (below) also show FE simulation results for a baseline design (solid line) as compared to the high damping design of the invention (dashed line). In particular, FIG. 3 shows out-of-plane displacement resulting from bending or torsion mode vibrations, where the vibration mode frequencies range from about 6,000 Hz to about 10,000 Hz. FIG. 4 shows off-track gain resulting from sway mode vibrations, where the vibration mode frequencies range from about 18,000 Hz to about 22,500 Hz.

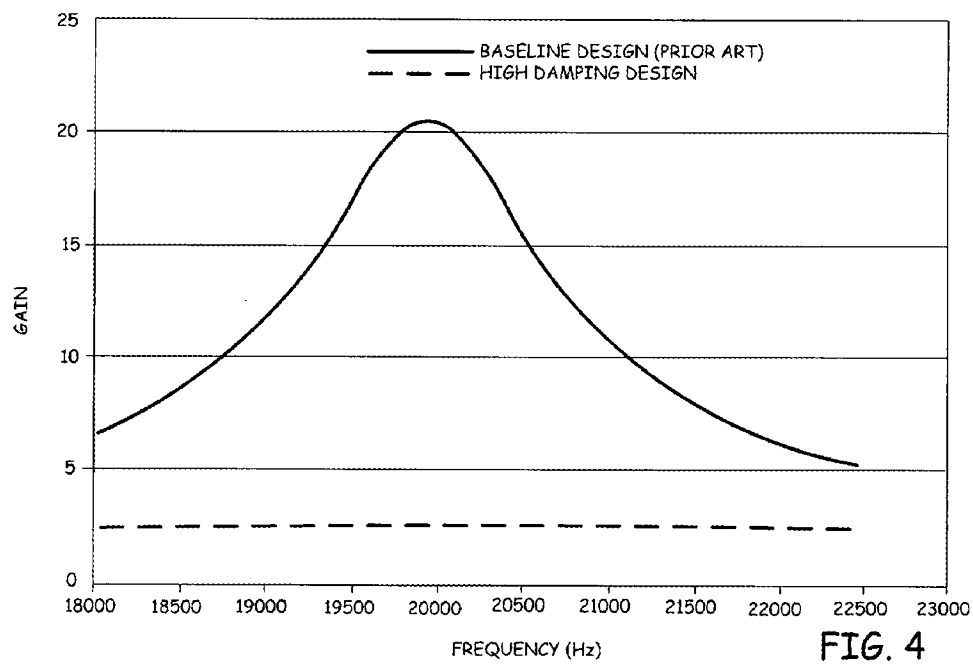

FIG. 4 of the Application

Claim 2 recites the head suspension assembly [16] of claim 1, where the first and second structural damping materials have a modulus of elasticity greater than approximately 30 GPa in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz. *Id.*, ¶¶ 34, 40; see also ¶¶ 4, 22, 26, 27; Table 1; FIGS. 3, 4.

Claim 3 recites the head suspension assembly [16] of claim 1, where the first and second structural damping materials are substantially identical in composition. *Id.*, ¶ 25.

Claim 5 recites the head suspension assembly [16] of claim 1, where the hinge component [34] applies a preload on the transducing head through the beam component [30]. *Id.*, ¶ 29.

Claim 6 recites the head suspension assembly [16] of claim 1, where the entire hinge component [34] is substantially made from the first structural damping material only. *Id.*, ¶¶ 20, 21, 30.

Claim 7 recites the head suspension assembly [16] of claim 1, where the entire gimbal component [36] is substantially made from the second structural damping material only. *Id.*, ¶¶ 24, 25, 30.

Claim 8 recites the head suspension assembly [16] of claim 1, where the hinge component [34] has no external structural damping material attached thereto. *Id.*, ¶¶ 30, 50.

Claim 9 recites the head suspension assembly [16] of claim 1, where the first structural damping material has a modulus of elasticity greater than approximately 50 GPa in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz. *Id.*, ¶¶ 34, 40.

Claim 10 recites head suspension assembly [16] of claim 1, where the second structural damping material has a modulus of elasticity greater than approximately 50 GPa in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz. *Id.*, ¶¶ 34, 40.

Claim 11 recites the head suspension assembly [16] of claim 1, where the first structural damping material is an alloy. *Id.*, ¶¶ 8, 41–43.

Claim 12 recites the head suspension assembly [16] of claim 1, where the first structural damping material is a laminate comprising a stainless steel layer and a damping material layer. *Id.*, ¶¶ 8, 41, 45–47.

Claim 14 recites the head suspension assembly [16] of claim 1, where at least one of the hinge component [34] and the gimbal component [36] is attached to the beam component [30] through an adhesive. *Id.*, ¶ 32.

Claim 15 recites the head suspension assembly [16] of claim 1, where at least one of the hinge component [34] and the gimbal component [36] is attached to the beam component [30] by welding. *Id.*, ¶¶ 32, 33.

Claim 26 recites the head suspension assembly [16] of claim 1, wherein the first structural damping material is a composite. *Id.*, ¶¶ 40, 41, 44, 45, 47.

Claim 32 recites the head suspension assembly [16] of claim 1, wherein the second structural damping material is an alloy. *Id.*, ¶¶ 8, 41–43.

Independent claim 16 recites a head suspension assembly [16] comprising a beam component [30] having a front end and a rear end, a hinge component [34] for connecting to an actuation arm [16] and a gimbal component [36] near the front end of the beam component [30] for connecting to a slider assembly [24] carrying a transducer. The hinge component [34] is separately made and attached to the rear end of the beam component [30]. *Id.*, ¶¶ 30, 32. The hinge component consists essentially of a first structural damping material having a modulus of elasticity greater than approximately 10 GPa and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz. *Id.*, ¶¶ 34, 40; see also ¶¶ 4, 22, 26, 27; Table 1; FIGS. 3, 4.

Claim 18 recites the head suspension assembly [16] of claim 16, where the first structural damping material is an alloy. *Id.*, ¶¶ 8, 41–43.

Claim 19 recites the head suspension assembly [16] of claim 16, where the gimbal component [36] comprises a second structural damping material having a modulus of elasticity greater than approximately 10 GPa and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz. *Id.*, ¶¶ 34, 40; see also ¶¶ 4, 22, 26, 27; Table 1; FIGS. 3, 4.

Claim 20 recites the head suspension assembly [16] of claim 19, wherein the first structural damping material and the second structural damping material are substantially identical in composition. *Id.*, ¶ 25.

Claim 27 recites the head suspension assembly [16] of claim 16, wherein the first structural damping material is a laminate comprising a stainless steel layer and a damping material layer. *Id.*, ¶¶ 8, 41, 45–47.

Claim 28 recites the head suspension assembly [16] of claim 16, wherein the first structural damping material is a composite. *Id.*, ¶¶ 40, 41, 44, 45, 47.

Claim 29 recites the head suspension assembly [16] of claim 19, wherein the second structural damping material is an alloy. *Id.*, ¶¶ 8, 41–43.

Claim 30 recites the head suspension assembly [16] of claim 19, wherein the second structural damping material is a laminate comprising a stainless steel layer and a damping material layer. *Id.*, ¶¶ 8, 41, 45–47.

Claim 31 recites the head suspension assembly of claim 19, wherein the second structural damping material is a composite. *Id.*, ¶¶ 40, 41, 44, 45, 47.

Grounds of Rejection to be Reviewed on Appeal

On this Appeal, the following grounds of rejection are to be reviewed, as raised in the final rejection mailed January 27, 2010.

> Whether claims 1–3, 5–12, 14–16, 18–20 and 26–32 are unpatentable under 35 U.S.C. § 103(a) as obvious over Arya et al., U.S. Patent No. 6,785,094 in view of Sutton et al., U.S. Patent No. 5,965,249 and Takagi et al., U.S. Pub. No. 2001/0008475.

Argument

In the final Office Action mailed January 27, 2010, claims 1–3, 5–12, 14–16, 18–20 and 26–32 were rejected under 35 U.S.C. § 103(a) as unpatentable (obvious) over Arya et al., U.S. Patent No. 6,785,094 (Arya), in view of Sutton et al., U.S. Patent No. 5,965,249 (Sutton) and Takagi et al., U.S. Pub. No. 2001/0008475 (Takagi), Appl. No. 09/793,410, abandoned. Based on the amendments of record and the arguments herein, each of claims 1–3, 5–12, 14–16, 18–20 and 26–32 is nonobvious under 35 U.S.C. § 103(a), and should be allowed.

1. Cited Art

a. Arya et al., U.S. Patent No. 6,785,094

Arya discloses a weld-free high performance laminate suspension assembly designed to carry a slider for a magnetic data storage device. The suspension assembly is formed from a composite laminate with five layers. A removal process such as partial chemical etching is used to define a mount plate, a hinge, a load beam and an electrical lead system. The structure may be also formed with a flexure gimbal system comprising the third, fourth and fifth layers. Arya, Abstract.

According to Arya, disk drive suspensions must be compliant to facilitate proper gram loading but should be relatively stiff in the direction parallel to the disk surface to prevent sway, with sufficient torsional stiffness to prevent rotational misalignment. The suspension must also have good dynamic characteristics to prevent unwanted vibration and flutter, and resonance at critical dynamic frequencies can induce unwanted torsion, sway and bending. Dynamic design considerations are particularly acute at increased recording density. Arya, col. 1, lines 46–54.

Arya teaches that disk drive suspensions have historically been formed by welding stainless steel elements to form a mount plate, hinge, flexure and load beam. The disadvantage is that welding requires additional fixturing and processing steps, introducing additional alignment tolerance. Welding also tends to reduce available real estate for piezoelectric milliactuators and other components, and thermal distortion at the weld points leads to flatness variations and flutter. Freely vibrating lengths between the weld points also contribute to dynamic flutter, and to mode gains at critical frequencies. Accordingly, what would be particularly desirable is a suspension that is substantially, if not completely, weld-free in its construction. Arya, col. 1, line 65 – col. 2, line 28.

Arya addresses these problems by forming a weld-free suspension from a composite laminate structure with five layers. A suitable process such as chemical etching is used to define a mount plate, a hinge, a load beam functional end, and an electrical lead system. A flexure gimbal system can also be formed, which may include a slider attachment area. Arya, col. 2, lines 32–59.

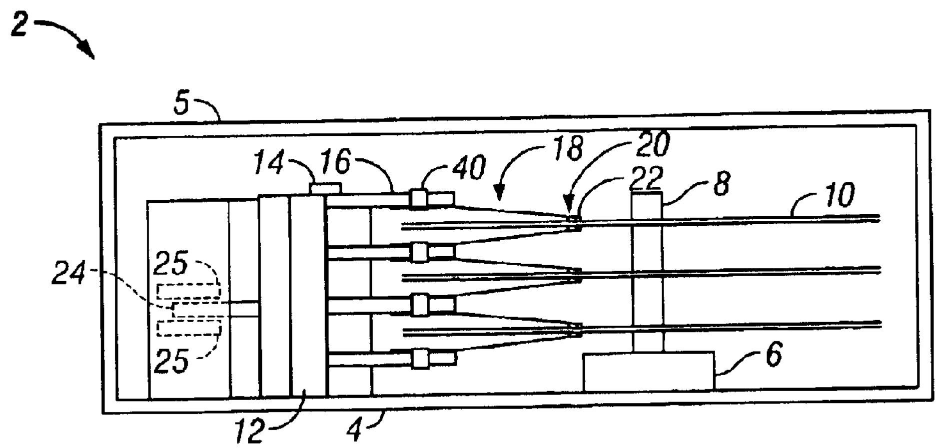

FIG. 2 of Arya

Arya's FIG. 2 (above) and FIG. 3 (below) show disk drive 2 with base casting 4, cover 5, spindle drive motor 6 and drive spindle 8 carrying a stacked array of magnetic disks 10. Actuator 12 rotates about pivot shaft 14, and actuator arms 16 carry flexible suspensions 18. Suspensions 18 support air-bearing sliders 20 with read/write transducers 22. Transducers 22 are positioned to read and write data by using voice coil motor winding 24 and motor magnets 25 to pivot actuator 12 about shaft 14. Actuator arms 16 sweep suspensions 18 across the disk surfaces, moving sliders 20 with transducers 22 generally radially from one concentric data track to another. Arya, col. 3, line 47 – col. 4, line 25.

Disks 10 spin up when disk drive 2 is powered on, causing an upward air-bearing force to develop between the disk surfaces and sliders 20. This is counteracted by a downward gram loading force provided by suspensions 18, enabling transducers 22 to fly above the disk surfaces. Data on disks 10 are read by transducers 22 and readback signals are sent to disk drive controller 26. Controller 26 also generates write signals, which are delivered to transducers 22 for writing data onto disks 10. Arya, col. 4, lines 26–52.

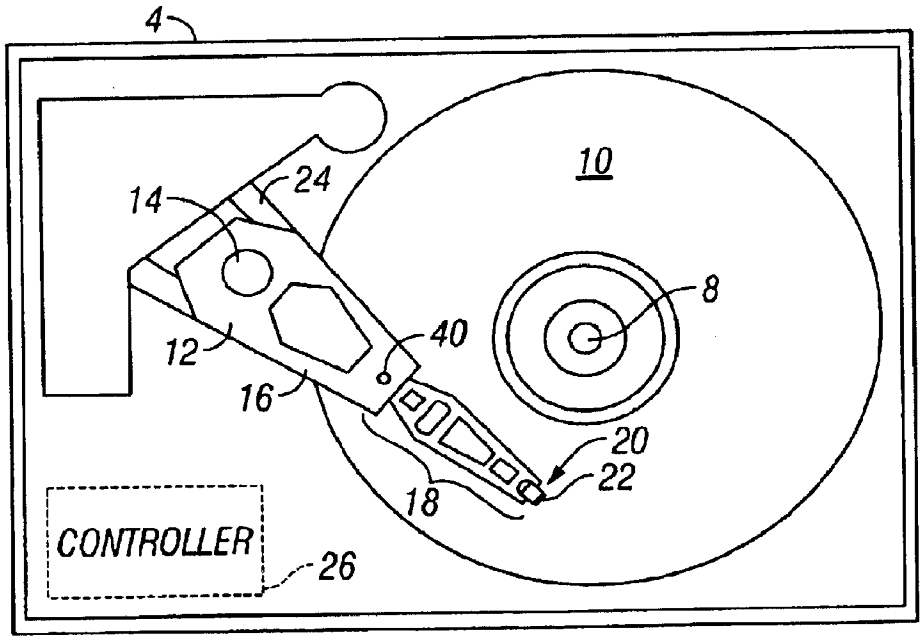

FIG. 3 of Arya

FIG. 6 (below) illustrates a method for manufacturing suspension 18 of FIGS. 2 and 3 from a composite laminate structure. The laminate includes first layer 32, second layer 34, third layer 36, fourth layer 38 and fifth layer 40. Arya, col. 4, lines 53–57.

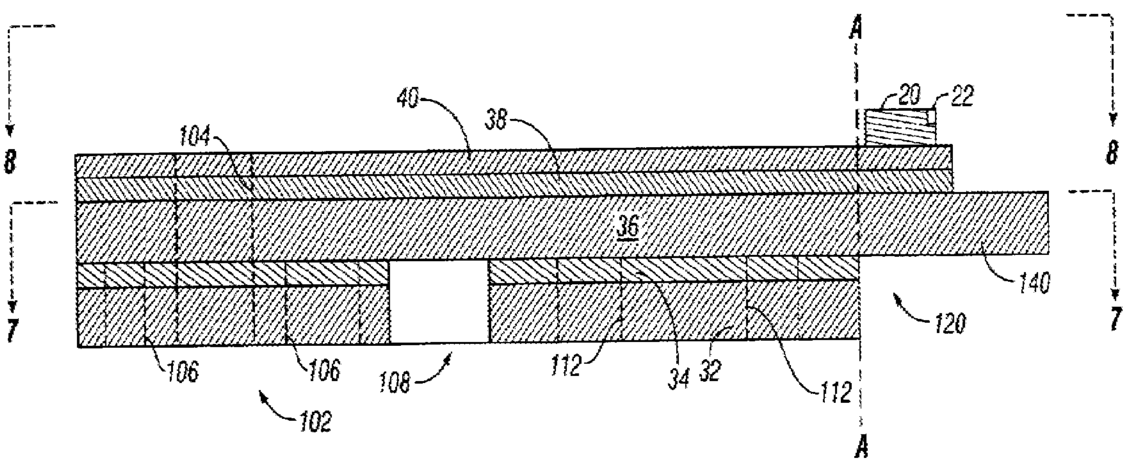

FIG. 6 of Arya

First layer 32 is a main load bearing layer for a mount plate and load beam functional end, and is preferably made from structural load bearing materials including stainless steel, copper, and glass/ceramic materials. Second layer 34 functions as an electrically insulating or conductive

(ground plane) layer or a damping layer, and is preferably made from electrically insulating or conducting materials or damping materials including polyimides, copper, aluminum, and viscoelastic polymers. Third layer 36 functions as a principal load bearing layer for a flexure, and is preferably made from structural load bearing materials including stainless steel and copper. Fourth layer 38 functions as a flexure insulative layer or a damping layer and is preferably made from electrically insulating materials or damping materials including polyimides and viscoelastic polymers. Fifth layer 40 functions as a flexure electrical lead layer and is preferably made from electrically conducting materials. The laminate structure can be formed from prefabricated, roll-formed sheets, from layers of polymeric material coated in liquid form onto adjacent metal layers, or using a conventional lamination process. Arya, col. 4, line 57 – col. 5, line 47.

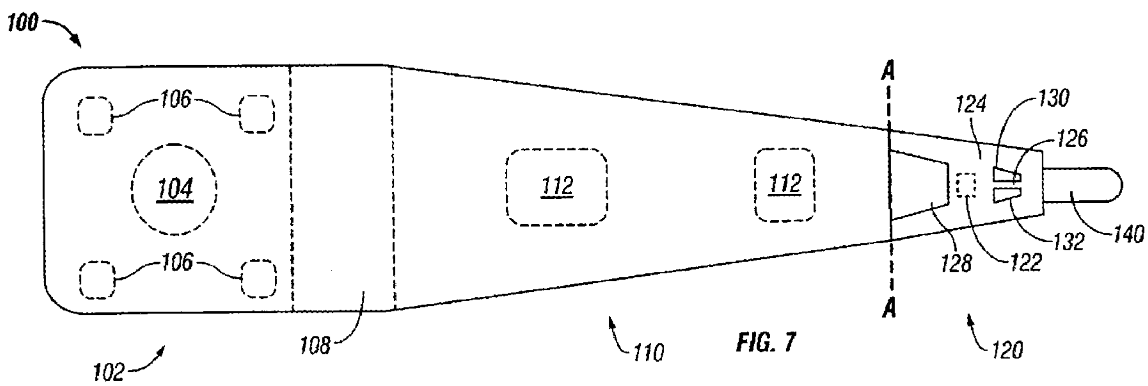

FIG. 7 of Arya

As shown in FIG. 7 (above) and FIG. 8 (below), suspension 100 includes mount plate 102 adapted for mounting to an actuator arm. Mount plate 102 acts as a rigid load bearing structure and comprises at least first layer 32, second layer 34 and third layer 36, which impart strength and rigidity. In one embodiment, first layer 32 is made from stainless steel, second layer 34 is made from a viscoelastic damping material and third layer 36 is made from stainless steel or copper. Mount plate 102 could also comprise one or both of fourth layer 38 and fifth layer 40, but these layers would not be relied on for structural properties and would usually be present as a matter of manufacturing convenience. The electrical lead system may also be carried across the mount plate 102. Arya, col. 5, line 48 – col. 6, line 15.

Attachment aperture 104 is formed by a process such as chemical etching, in order to attach suspension 100 to an actuator arm using a swage or rivet connector. Mass reducing pockets 106 are formed by removing all or part of first layer 32 at selected locations. Hinge 108 is formed adjacent to mount plate 102 by removing all or part of first layer 32 and second layer 34. Third through fifth layers 36, 38, 40 are preferably left intact, with third layer 36 acting as a load bearing member, preferably made from stainless steel or copper with good bending compliance to facilitate gram loading. Fourth and fifth layers 38 and 40 provide electrical lead connections to and from the transducer 22. Arya, col. 6, lines 16–40.

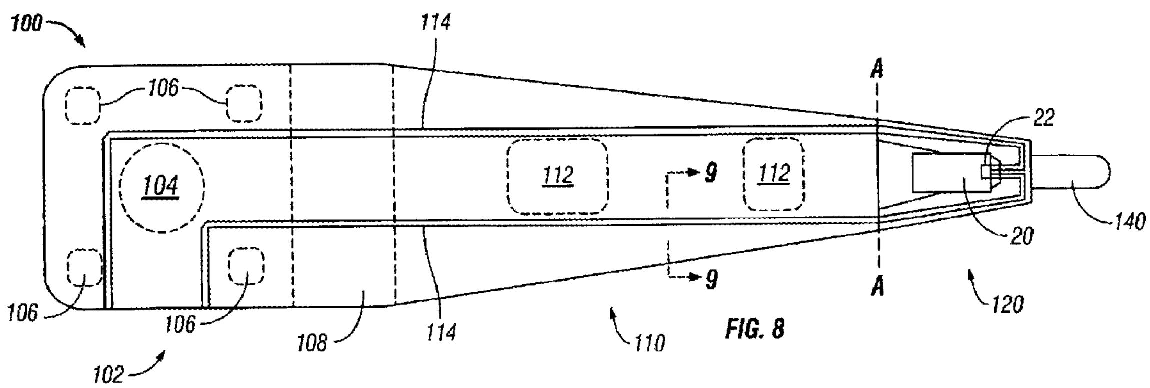

FIG. 8 of Arya

Load beam functional end 110 is formed adjacent to hinge 108, opposite mount plate 102, and acts as a stiff load bearing structure that carries electrical lead elements. Functional end 110 comprises at least third layer 36, fourth layer 38 and fifth layer 40, and preferably first and second layers 32 and 34. Third through fifth layers 36, 38, 40 of functional end 110 support slider 20 and carry electrical signals to and from transducer 22. In some embodiments, third layer 36 is made from stainless steel, fourth layer 38 is made from a polyimide material and fifth layer 36 is made from copper.[1] Third layer 36 could also be made from copper to define a ground plane. Arya, col. 6, lines 41–65.

First layer 32 is preferably made from stainless steel, copper or a glass/ceramic material and imparts additional structural stiffness. Second layer 34 is preferably made from aluminum, copper, a polyimide or a viscoelastic damping material and provides additional properties

[1] Sic. Presumably refers to fifth layer 40.

according to the material selected. Functional end 110 may also be configured with one or more mass reducing pockets 112 formed by removing all or part of third layer 36 or first layer 32 (if present) at selected locations, using a process such as partial chemical etching. Arya, col. 6, line 66 – col. 7, line 14.

An electrical lead system is formed on suspension 100 by selectively patterning fifth layer 40, which is preferably made from an electrical conductor such as copper, to define electrical lead traces 114 connecting to transducer 22. Traces 114 are defined by first portions 116 of fifth layer 40, where partial chemical etching can be used to remove second portions 118 and electrically isolate first portions 116. Arya, col. 6, lines 15–25.

Functional end 110 can also be configured with flexure gimbal system 120, which extends to the right of line A–A and carries slider 20 and transducer 22 in a gimbaled support arrangement. Flexure gimbal system 120 is created by selectively patterning third layer 36 to form structures including slider attachment area 122 at the intersection of gimbaling flexure members 124 and 126. Arya, col. 6, lines 26–36.

Slider attachment area 122 and gimbaling flexure members 124 and 126 can be formed using three partial etch processes. Initially, suspension 100 is partially etched to remove portions of first and second layers 32 and 34 to the right of line A–A, clearing space for etching flexure gimbal system 120 in third layer 36. Suspension 100 is then etched to remove portions of fourth and fifth layers 38 and 40 to the right of the line A–A, which are not associated with electrical lead traces 114. Third layer 36 is then etched to create openings 128, 130 and 132, defining gimbaling flexure members 124 and 126 as a T-shaped structure in third layer 36. Arya, col. 6, lines 37–55.

Slider attachment area 122 at the intersection of flexure members 124 and 126 provides a mounting structure for supporting slider 20 in a gimbaled arrangement. When flexed, flexure members 124 and 126 tend to pivot slider attachment area 122, allowing the slider 20 to pitch and roll during drive operation. Load/unload tang 140 could also be formed at the right hand end of suspension 100, preferably after formation of flexure gimbal system 120. Arya, col. 6, lines 56–67.

b. Sutton et al., U.S. Patent No. 5,965,249

Sutton discloses a composite damping material achieved through the entrapment of highly viscous damping fluids within the pores of a porous material such as an expanded polymer, felt, foam, fabric or metal. Sutton teaches applications including airplanes, automobiles, space structures, machine tools, sporting goods, disk drive components, electrical components and cables. Sutton, Abstract.

According to Sutton, damping losses are quantified in terms of Young's moduli or dynamic shear moduli. The dynamic storage modulus is proportional to the amplitude of the stress response resulting from a sinusoidal strain, where the strain may be either shear or elongational depending on whether the shear modulus or Young's modulus is desired. The loss modulus is proportional to the amplitude of the stress response resulting from the sinusoidal strain rate, and the ratio of the dynamic shear loss modulus to the dynamic shear storage modulus (or dynamic Young's loss modulus to dynamic Young's storage modulus), is referred to as tan(δ), as defined a particular oscillation frequency. The magnitude of the loss modulus quantifies viscous-like resistance to deformation while tan(δ) quantifies the relative magnitude of this resistance to elastic response. Sutton, col. 1, lines 35–54.

Sutton also teaches that improved damping characteristics are commonly achieved at the expense of other desired properties, and that fundamental limitations exist on the damping performance of articles with good mechanical stability. In particular, Sutton teaches that internal loss is correlated with cold flow, and increasing the damping properties beyond a given point often leads to the inability to maintain shape, even under the influence of gravity. Sutton, col. 2, line 58 – col. 3, line 5.

Sutton's invention relates to a vibration damping article comprised of a porous scaffold to provide mechanical stability and a material which is mechanically unstable with respect to cold flow but which provides high viscous loss characteristics. Sutton, col. 2, lines 5–9. In particular, Sutton's objectives are met by a composite film, sheet or tube comprised of a high dynamic loss viscoelastic material filled into the pores of a porous polymer, ceramic, glass or metal substrate. Porous polymer substrates include fluorinated organic polymers, polyamides, polyolefins,

polyurethanes, polyethylene, PVC and cellulose acetate. Porous ceramics include porous sintered silicas, carbides and aluminas. Porous metal substrates include sintered porous aluminum and stainless steel. The viscoelastic components include polymer resins, fluorocarbons, polyurethanes, acrylics, silicones, polyisobutylenes and waxes. Sutton, col. 4, line 33 – col. 5, line 40.

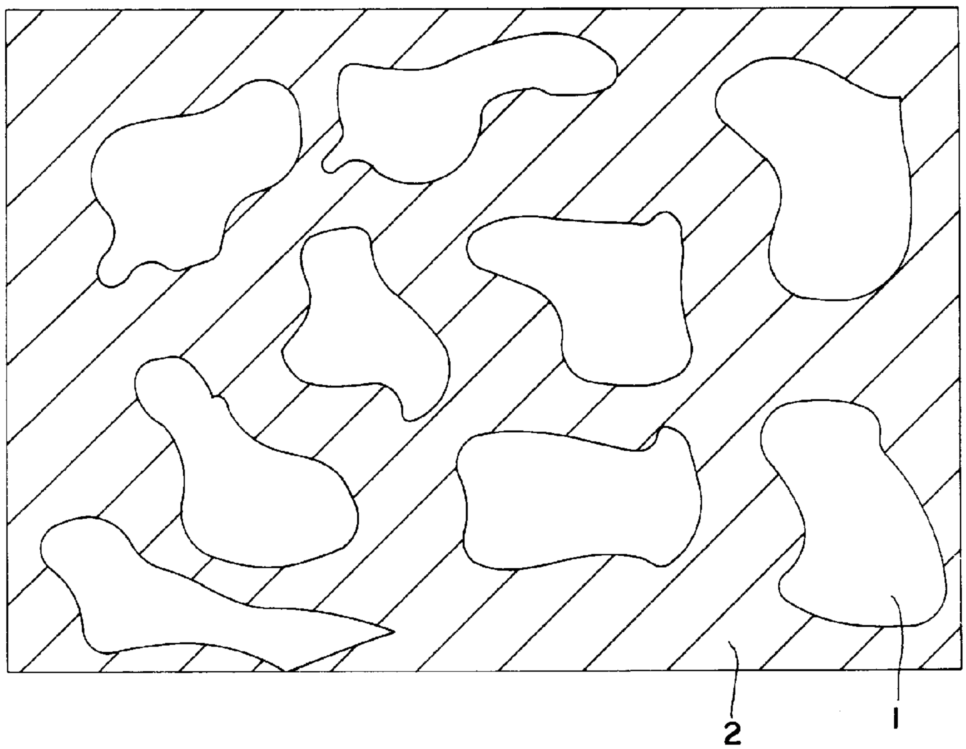

FIG. 1 of Sutton

In one embodiment, viscoelastic damping material 1 is entrapped within the closed cells of porous material 2 as shown in FIG. 1 (above). Provided diffusion is negligible and porous backbone 2 is stable, damping component 1 is fully entrapped within the composite, preventing cold flow. Provided the closed cell structure is comprised of a sufficiently rigid porous material 2, deformation of the composite structure results in deformation of damping component 1, and the desired damping response is achieved. Sutton, col. 10, lines 35–48.

In open cell porous substrates, binding is accomplished by a number of means. In one such embodiment, small drop 3 of a viscoelastic damping fluid 1 is constrained within scaffold 4 of substrate material 2, as shown in FIG. 2 (below). In this example the composite material is stable with respect to cold flow by virtue of the mechanical integrity of the porous matrix, and capillary

forces trap damping material 1 within porous substrate 2 because the convoluted paths to the interior of scaffold 4 require the surface of drop 3 to be broken. Viscous forces and other mechanisms also combine to bind viscoelastic damping materials 1 within various porous substrates 2, preferably with a substantially uniform distribution. Sutton, col. 11, line 28 – col. 12, line 14.

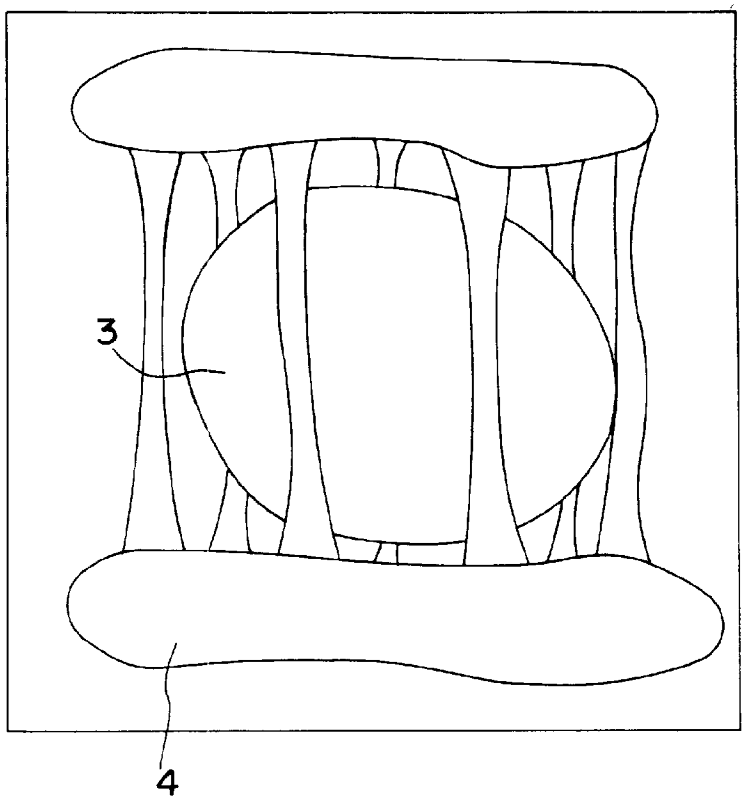

FIG. 2 of Sutton

Sutton teaches dynamic mechanical analysis (DMA) to define the composite material properties by applying a swatch of the material to a steel beam and subjecting the beam to three-point bending oscillations. The resulting dynamic forces are decomposed into in-phase and out-of-phase components, and these components are used to calculate effective values for the dynamic loss, storage moduli and tan(δ). Sutton, col. 15, line 65 – col. 17, line 42.

According to Sutton, calculating the dynamic loss and storage moduli is trivial for simple beam geometry but this is not the case for a composite system comprised of a beam and a bonded swatch of damping material. In particular, the effective values of the dynamic loss moduli and tan(δ) apply to the composite beam-plus-damping-material system, and it is not possible to

rigorously determine values for the individual beam and damping material components without a complex strain analysis of both the beam and the swatch. Sutton, col. 18, lines 16–24.

Instead, Sutton defines the effective dynamic loss moduli and tan(δ) to represent the response that would be obtained if a pure beam of material yielded exactly the same force response as that of the (actual) steel beam combined with the (actual) damping swatch. These effective quantities still differentiate among different damping materials, and materials with superior damping performance will yield higher effective dynamic loss moduli and effective tan(δ). Sutton, col. 18, lines 25–67. While the effective dynamic loss moduli and effective tan(δ) allow the relative merits of different standard swatches to be determined, they are not representative of real quantities for the components of the system. Sutton, col. 18, lines 39–42.

Sutton also teaches direct determination of the dynamic loss and storage moduli by cutting a sample swatch of the composite damping material and subjecting it to sinusoidal oscillations. Sutton, col. 19, line 18 – col. 20, line 39. Sutton also teaches droop tests to quantify the instability of the viscoelastic damping material with respect to cold flow. Sutton, col. 20, line 40 – col. 21, line 61.

In both the composite beam and direct dynamic tests, the DMA oscillation frequencies range from about 1–100 radians/second (about 1/6 Hz to about 16 Hz). Sutton, col. 19, lines 6–8; col. 20, lines 33–35. Typical applications, however, require knowledge of damping performance at frequencies beyond 22 kHz, and it is desirable to compare materials across a broader range of acoustic frequencies. But dynamic testing is difficult and costly at these frequencies, and Sutton employs time-temperature superposition instead. Sutton, col. 21, line 65 – col. 22, line 14.

Time-temperature superposition is based on the assumption that frequency-dependent mechanical properties of polymeric materials scale in a specific fashion with temperature, so that dynamic processes at a particular reference temperature are related to those at higher frequency but lower temperature. For this analysis, Sutton assumes that the effective dynamic and storage moduli scale with an empirical constant α. Sutton then constructs time-temperature master curves by making measurements at different temperatures of –10, 0, 10, 20, 30 and 40° C, across the range of frequencies achievable with the DMA instrumentation, and adjusts the scaling constant until an

appropriate continuous (or near continuous) curve is obtained. According to Sutton, the resulting master curves represent the expected dynamic moduli across a very broad range of frequencies, at particular reference temperatures T_{ref}, for example 20° C. Sutton, col. 7, line 1 – col. 9, line 42; col. 22, line 15 – col. 23, line 4; col. 25, lines 1–63; col. 28, line 56 – col. 29, line 53.

Sutton notes that the applicability of time-temperature superposition to three-point bending data for a composite system is less obvious than for fiber-film data, and states that the superposition should not be applied to the storage modulus for the steel beam/composite swatch system. Nonetheless Sutton asserts that the techniques yield accurate time-temperature master curves for the effective variables, and defines an effective tan(δ) for relative comparisons based on the effective storage modulus at a particular scaled frequency f_{scaled} and reference temperature T_{ref}, using the derived dynamic loss master curve at frequency f. Sutton, col. 23, lines 5–42.

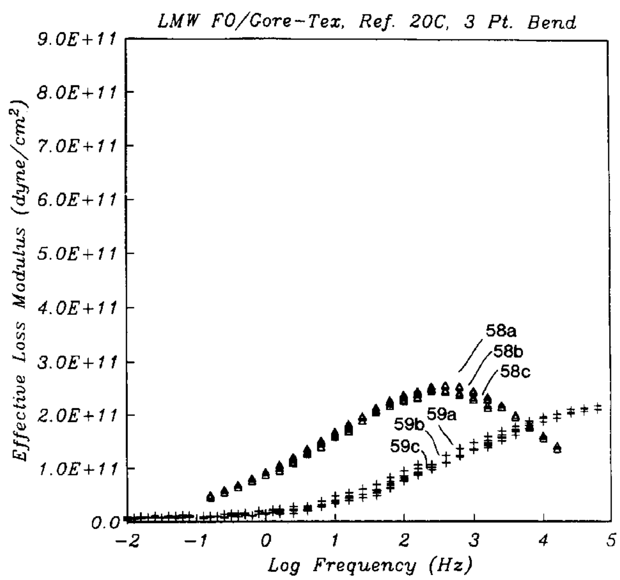

FIG. 18

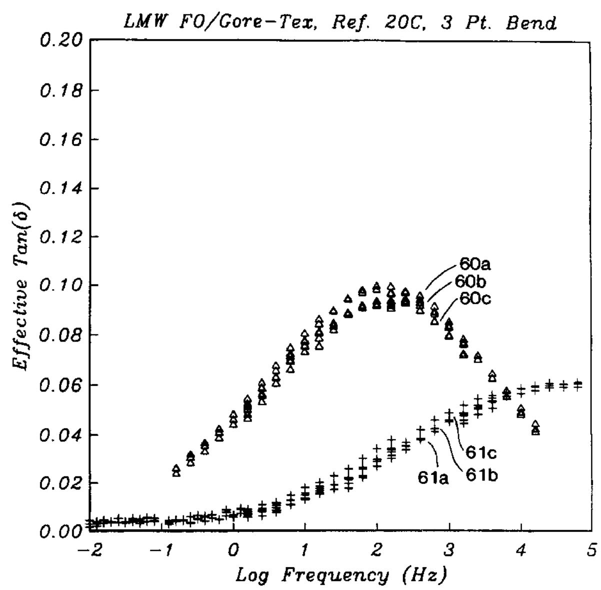

FIG. 19

FIGS. 18 (left) and 19 (right) of Sutton

Results for various sample materials and scaled frequencies are presented in FIGS. 18 and 19 (above) and in FIGS. 20–29 and 31–38 (below). Again, these effective loss moduli and effective tan(δ) data provide for appropriate relative comparison, but do not represent real quantities for the actual components of the system. Sutton, col. 18, lines 39–42.

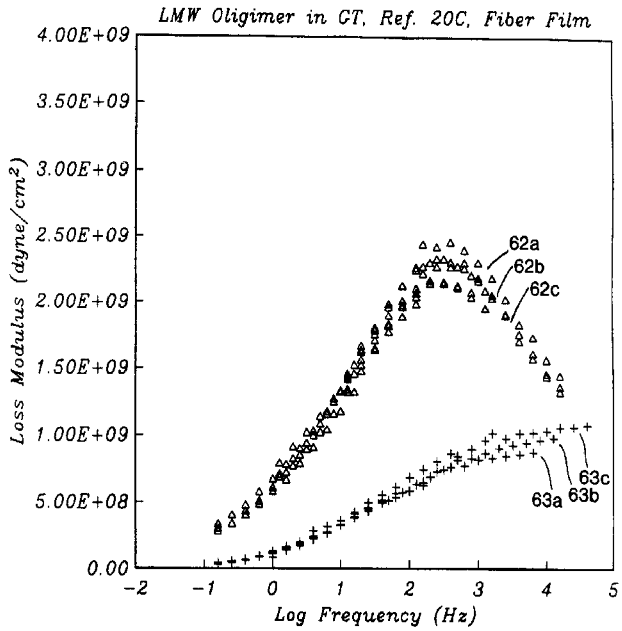

FIG. 20

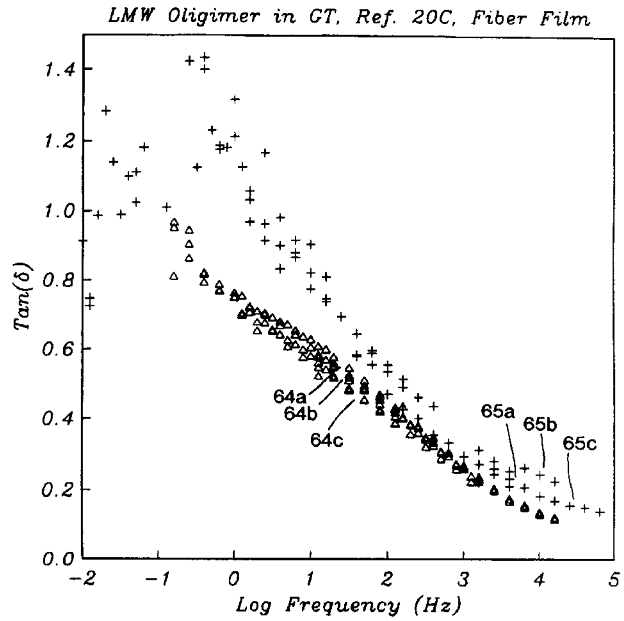

FIG. 21

FIGS. 20 (left) and 21 (right) of Sutton

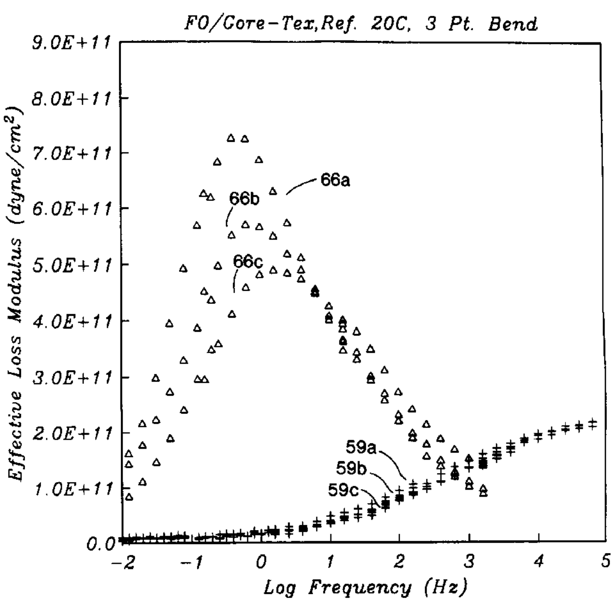

FIG. 22

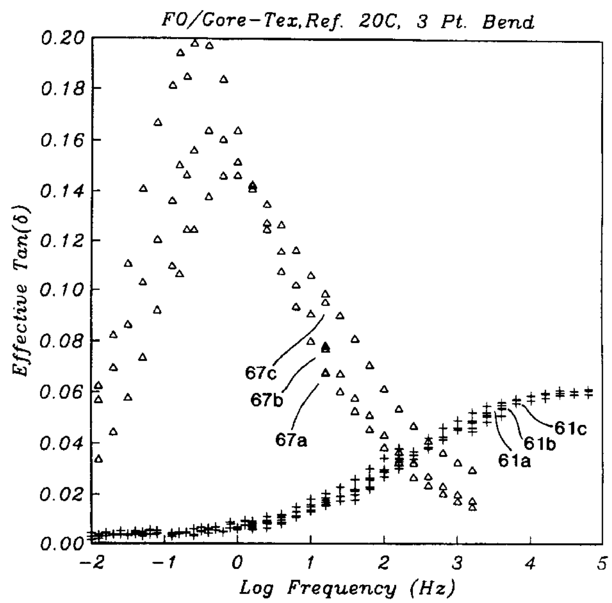

FIG. 23

FIGS. 22 (left) and 23 (right) of Sutton

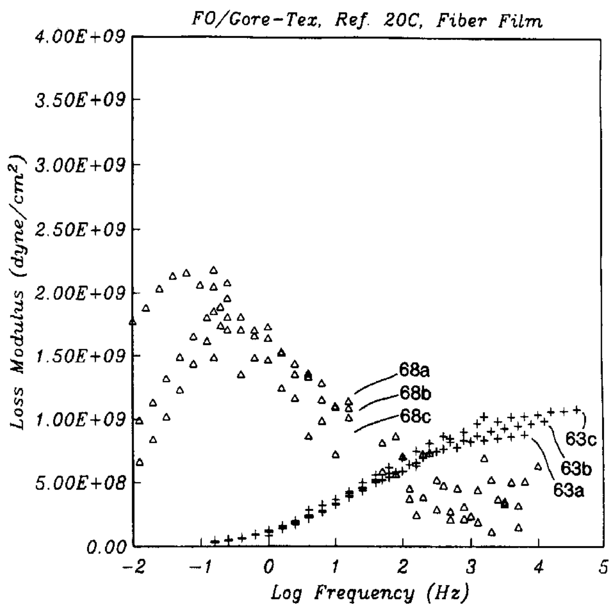

FIG. 24

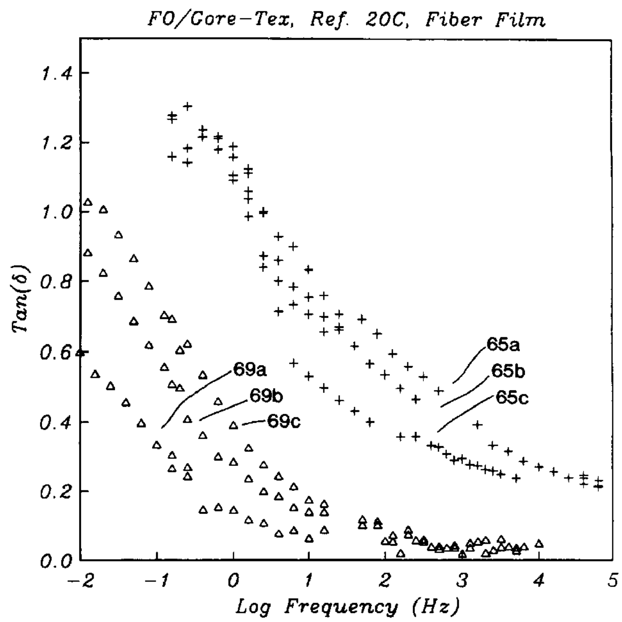

FIG. 25

FIGS. 24 (left) and 25 (right) of Sutton

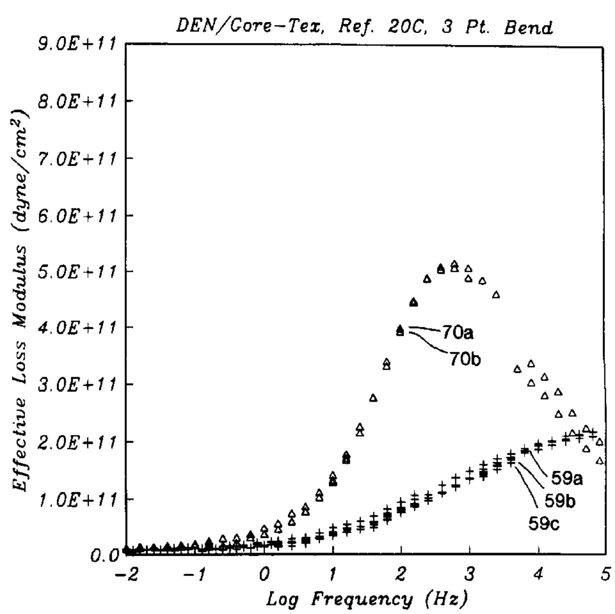

FIG. 26

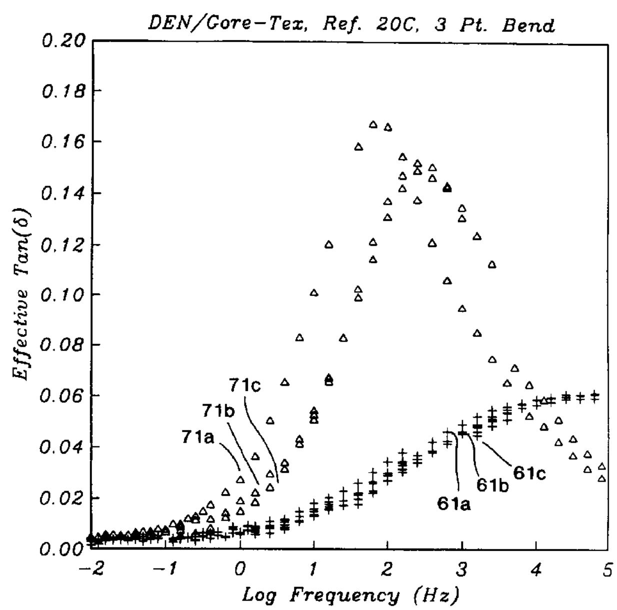

FIG. 27

FIGS. 26 (left) and 27 (right) of Sutton

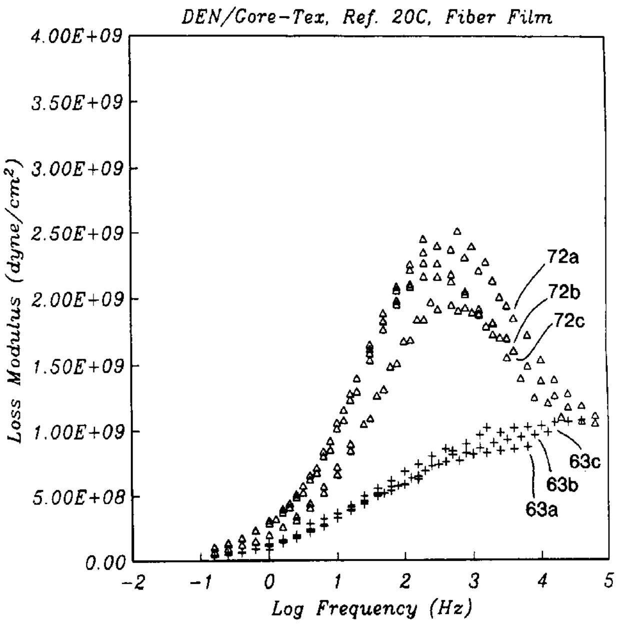

FIG. 28

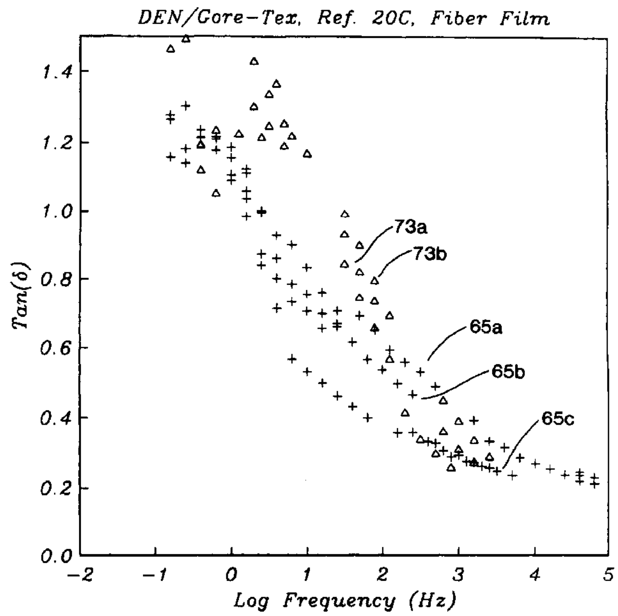

FIG. 29

FIGS. 28 (left) and 29 (right) of Sutton

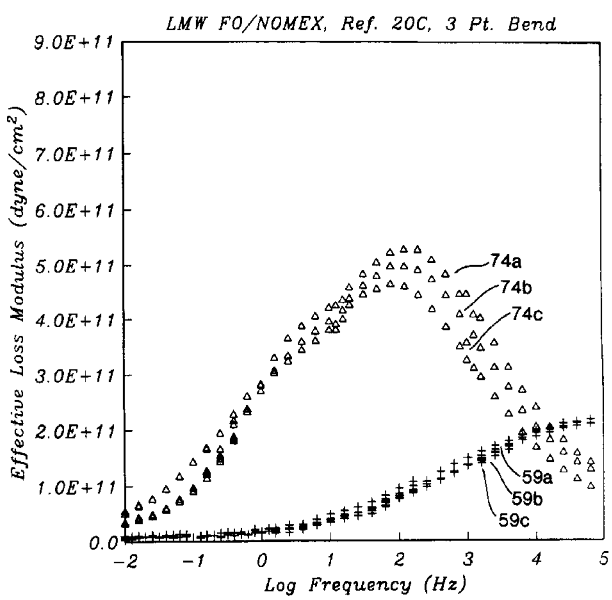

FIG. 31

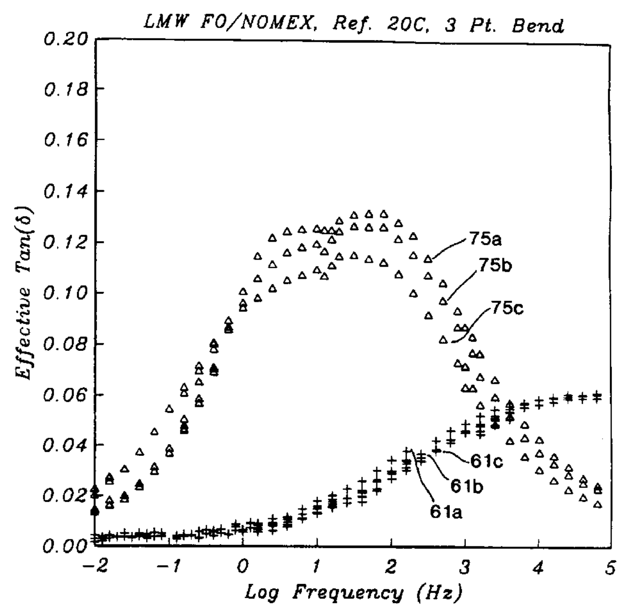

FIG. 32

FIGS. 31 (left) and 32 (right) of Sutton

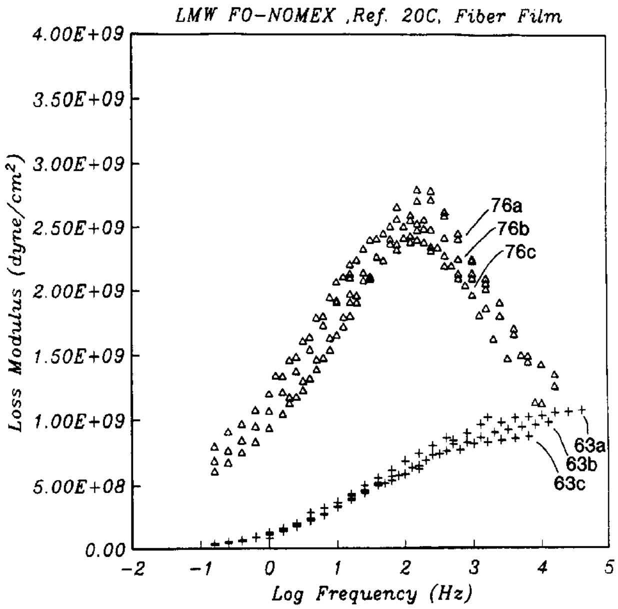

FIG. 33

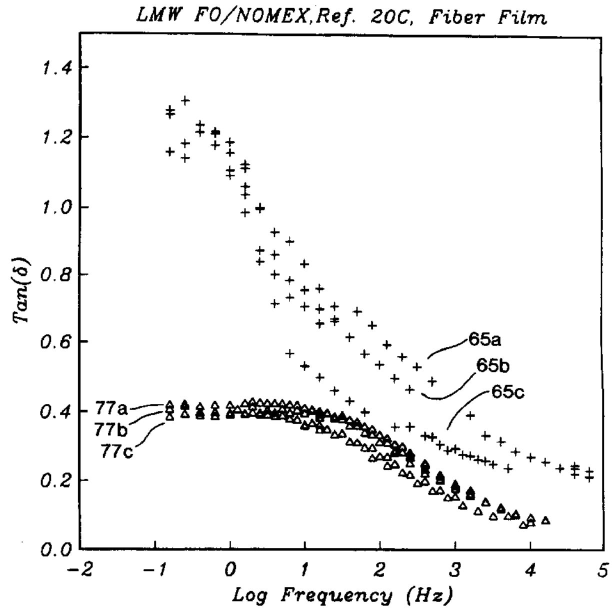

FIG. 34

FIGS. 33 (left) and 34 (right) of Sutton

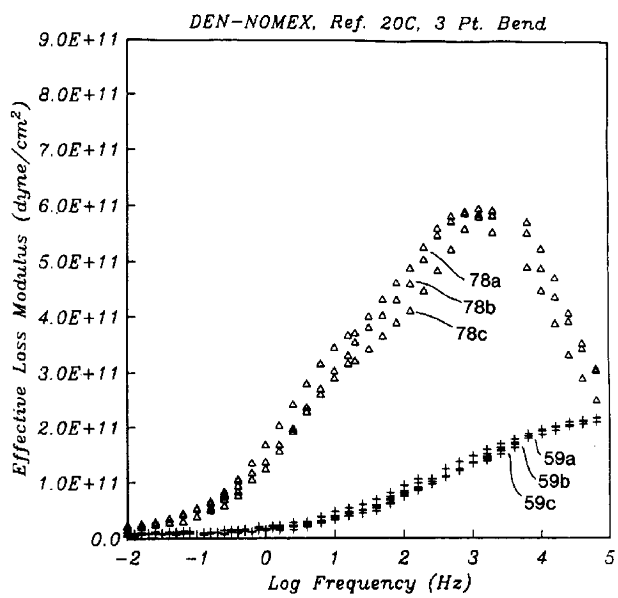

FIG. 35

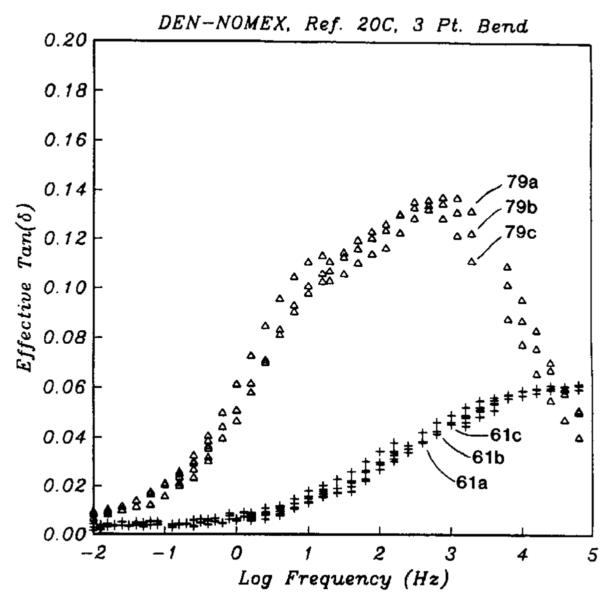

FIG. 36

FIGS. 35 (left) and 36 (right) of Sutton

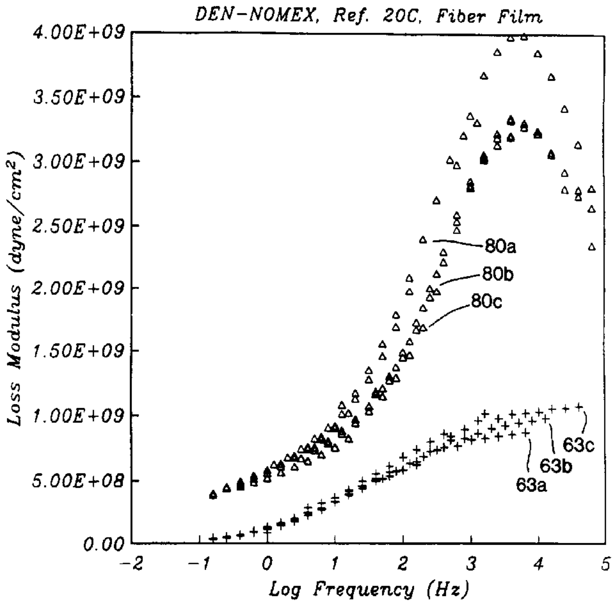

FIG. 37

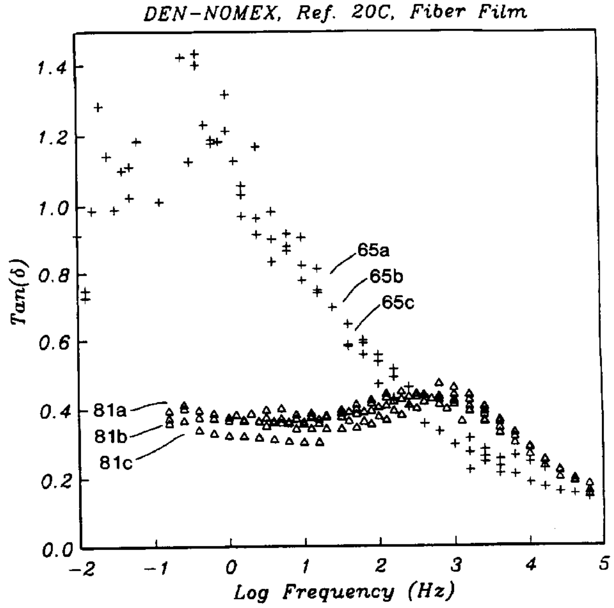

FIG. 38

FIGS. 37 (left) and 38 (right) of Sutton

c. Takagi et al., U.S. Pub. No. 2001/0008475

Takagi discloses a disc drive suspension comprising a base plate, a load beam attached to the base plate and a flexure attached to the load beam. The load beam includes a rigid body portion formed of a thick plate, independent of the base plate, and a spring portion formed of a thin sheet spring member, independent of the rigid body portion, where the spring member connects the rigid body portion and the base plate. The spring member is thinner than the rigid body portion, has a lower spring constant, and is more flexible than the rigid body portion. Takagi, Abstract.

The object of the Takagi invention is a high-performance disc drive suspension and a manufacturing method therefore. In one example, a thick plate is used for the rigid body portion of the load beam and thin rolled steel sheet is used for the spring. Takagi, ¶¶ 13–15. In another example, stainless steel is used for the rigid body portion, or an alloy of a light metal such as titanium or aluminum, or a synthetic resin for further weight reduction and higher stiffness. Alternatively, the load beam and other components are laminated with an aluminum-based metal, or with titanium or another metal or alloy, for example by cladding or bonding. Takagi, ¶ 16.

Forming the rigid body portion from a thick plate enhances stiffness and reduces air resistance without forming bent edges or ribs on the load beam, lessening turbulence and fluttering. Since the rigid body portion and the spring are separate components, the rigid body portion can be formed of a softer material. Takagi, ¶¶ 17–18.

Preferably, the rigid body portion is formed of a light metal or synthetic resin. If formed of a material with low specific gravity, the weight of the load beam is reduced and its frequency and vibration characteristics are improved. Weight is further reduced by using low-gravity material for both the load beam and the base plate, and the flexure and the spring portion are preferably formed of an integral metal sheet, reducing the number of components and improving positioning accuracy for the flexure and spring. Takagi, ¶¶ 19, 20.

A method for manufacturing the suspension utilizes a semi-finished suspension product including a base plate, a rigid body portion and a pair of connecting portions connecting the base plate and the rigid body portion. The distance between the connecting portions is greater than the width of the spring member, which is formed independently and fixed to the base plate and the rigid body portion. The connecting portions project from opposite sides of the spring member, and are cut off from the base plate and rigid body portion. A common material can be used for the base plate and the rigid body portion, reducing the number of components and increasing positional accuracy. Takagi, ¶ 21.

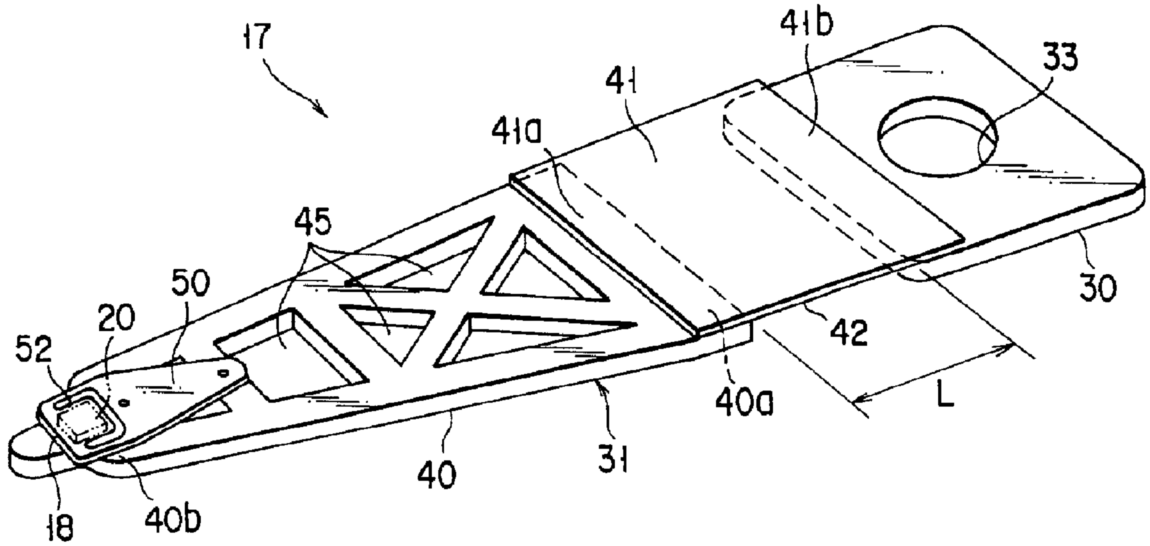

FIG. 1 of Takagi

As shown in FIG. 1 (above), suspension 17 includes base plate 30 and load beam 31. Base plate 30 has circular hole 33 for inserting a boss portion of actuator arm 16 (FIG. 3, below). Each load beam 31 is provided with rigid body portion 40, which is independent of base plate 30. Spring portion 42 is formed of spring member 41 fixed to body portion 40, where length L of spring member 41 functions as spring portion 42 and rigid body portion 40 is thicker than spring member 41. In addition, spring portion 42 has a smaller spring constant and is more flexible than rigid body portion 40, which is formed of a lightweight alloy with high stiffness such as an aluminum alloy, and is penetrated by apertures 45. Takagi, ¶¶ 38–42.

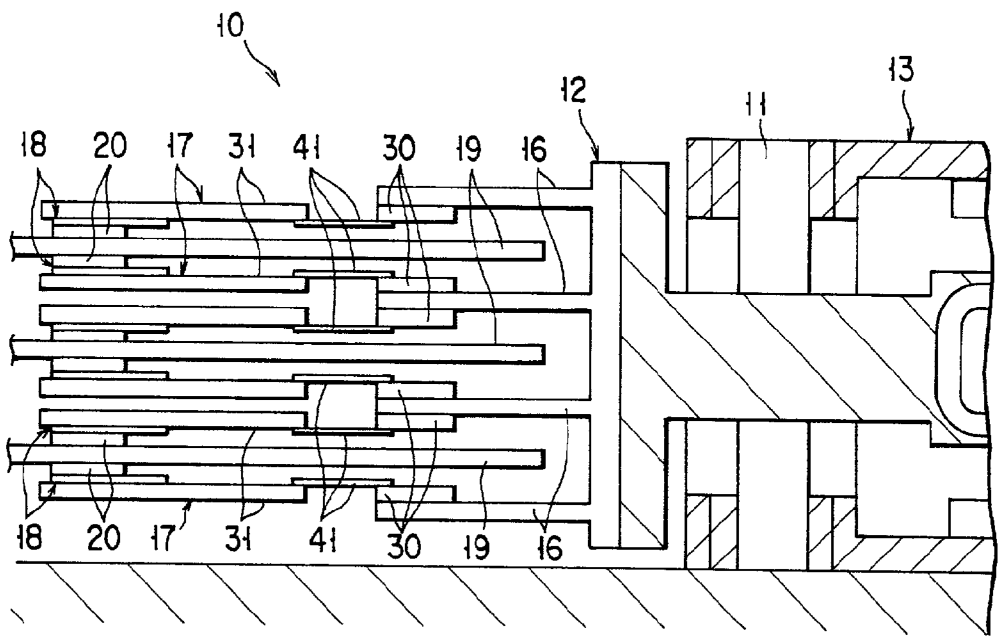

FIG. 3 of Takagi

Apertures 45 may be replaced with recesses formed by partially reducing the thickness of rigid body portion 40, for example by etching. Load beam 31 may be formed of a light metal (lower in specific gravity than iron) such as a titanium or aluminum alloy, or a synthetic resin. Low-gravity materials reduce the weight of load beam 31, and improve its frequency and vibration characteristics. Load beam 31 may also be a laminated member made up of aluminum alloy and stainless steel plates. Takagi, ¶ 43.

In one example plate-like spring member 41 is formed of a springy rolled stainless steel sheet. End portions 41a and 41b of spring member 41 are fixed to end portion 40a of rigid body

portion 40 and base plate 30 by laser welding or the like. Spring member 41 may also be fixed to rigid body portion 40 of load beam 31 with an adhesive. When body portion 40 is formed of a synthetic resin, spring member 41 may be fixed to body portion 40 by in-mold forming. Takagi, ¶¶ 44, 45.

Flexure 50 comprises a very thin plate spring attached to rigid body portion 40. In one example, flexure 50 is formed of a rolled stainless steel sheet and fixed to load beam 31 by laser welding or the like. Slider 20 with head portion 18 is mounted on flexure 50. Takagi, ¶ 46.

Rigid body portion 40 and spring portion 42 are separate components, with suitable materials and thicknesses to provide high stiffness for body portion 40 and a low spring constant for spring portion 42, using a high-accuracy rolled sheet for spring member 41. A thick plate can be used for load beam 31, such that load beam 31 can be shaped to reduce turbulence and enhance stiffness. Takagi, ¶¶ 47, 48.

In a second embodiment, spring portion 42 includes bent portions formed by bending the longitudinally intermediate portion of spring member 41. Takagi, ¶ 49. In a third embodiment, an aperture is formed in the central portion of spring member 41. Takagi, ¶ 50. In a fourth embodiment, the base plate is of the so-called arm type and the flexure 50C has a wiring board including a metal substrate 65, with an insulating layer and conductive lines connected to terminals on head portion 18 and the base plate. Takagi, ¶ 51.

In a fifth embodiment, the distance between the connecting portions is greater than the width of spring member 41. Takagi, ¶¶ 52–54. In a sixth embodiment, spring portion 42 and flexure 50 are formed from one platelike spring member. Takagi, ¶ 55. In a seventh embodiment, spring portion 42 and a portion to be put on base plate 30 are formed integrally on a platelike spring member constituting a flexure with a wiring board. Takagi, ¶ 56.

In an eighth embodiment ("SUS/Al Clad-beam"), rigid body portion 40 of load beam 31 is a laminated member made up of an aluminum alloy plate having a thickness of 100 μm and a stainless steel plate and having a thickness of 30 μm, and base plate 30 is a laminated member made up of a light alloy plate such as an aluminum alloy plate and a stainless steel plate. In this embodiment, spring member 41 is formed of stainless steel and is laser-welded to the stainless steel

plate. Takagi, ¶ 57. In a ninth embodiment, the aluminum alloy and stainless steel plate thickness are not described. Spring member 42 is part of the stainless steel plate and base plate 30 is made of a light alloy such as an aluminum alloy, which is overlaid on the stainless steel plate. Takagi, ¶ 58.

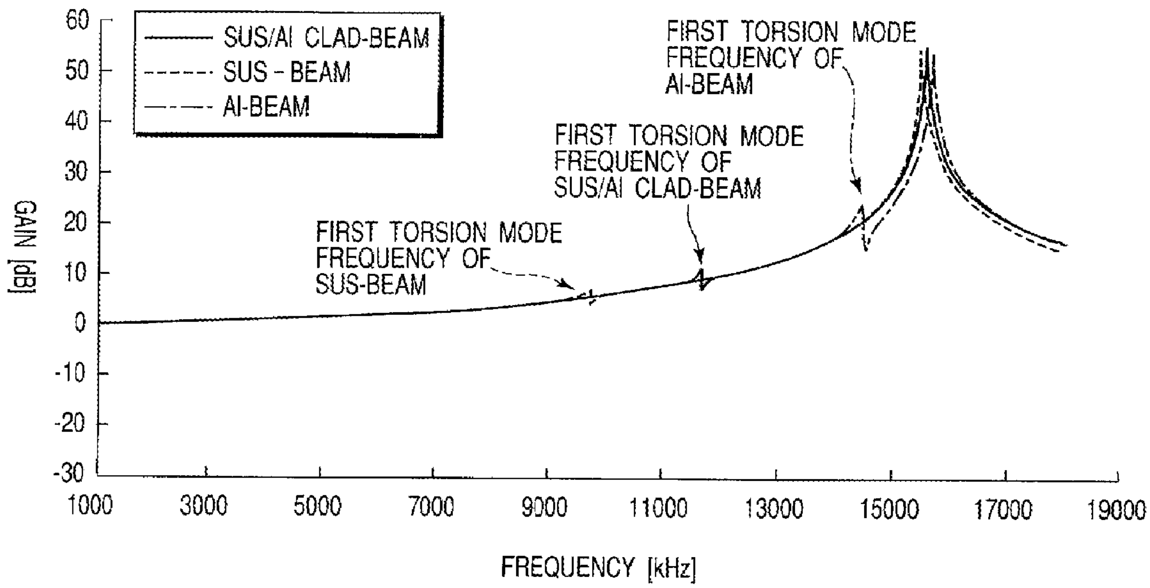

FIG. 13 of Takagi

FIG. 13 of Takagi (above) shows that the first torsion mode resonance frequency of the SUS/Al Clad-beam is higher than that of a beam formed only of stainless steel (SUS-beam). In addition, the SUS/Al Clad-beam is lighter in weight than a 100 μm thick SUS-beam, and has reliable frequency and vibration characteristics. A 160 μm Al beam formed of aluminum alone may show satisfactory frequency characteristics, but such a beam cannot easily be laser-welded to a flexure or base plate formed of stainless steel. Takagi, ¶ 59.

2. <u>Response to Claim Rejections</u>

On pages 2–18 of the Office Action, claims 1–3, 5–12, 14–16, 18–20 and 26–32 are rejected as unpatentable (obvious) over Arya in view of Sutton and Takagi. In making these rejections, the Office Action relies on Arya for teaching a suspension assembly, on Sutton for teaching structural damping materials, and on Takagi for teaching a hinge or gimbal that is separately made and attached to the beam. (Office Action, pp. 2, 3, 6, 7.) Sutton, however, does not disclose, teach or suggest the claimed damping capacities and elastic moduli in the recited frequency range. In addition, Arya teaches away from the proposed combination with Takagi, and away from Applicants' invention as claimed. Each of claims 1–3, 5–12, 14–16, 18–20 and 26–32 is nonobvious under 35 U.S.C. § 103(a), and should be allowed.

a. Claims 1, 16 and 19 are nonobvious under 35 U.S.C. § 103(a) because the recited ranges of elastic modulus and damping capacity would not have been obvious over the prior art.

Claim 1 and claim 16 each recite a suspension assembly having a beam component, a hinge component for connecting to an actuation arm and a gimbal component for carrying a transducer. Claim 1 specifies that the hinge component and the gimbal component each comprise a structural damping material having a modulus of elasticity greater than approximately 10 GPa and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz. Claim 16 specifies that the hinge component consists essentially of a structural damping material having the recited damping capacity and modulus of elasticity. Claim 19, which depends from claim 16, specifies that the gimbal component comprises a structural damping material having the recited damping capacity and modulus of elasticity.

In rejecting these claims, the Office Action asserts that (I) Sutton's materials have "damping capacities and moduli of elasticity within the claimed ranges;" (II) a person of ordinary skill would have been motivated to "optimize structural damping materials for use within [an] ever-higher frequency range;" and (III) "when the mere difference between the claimed invention and the prior art is some range, variable or other numeric limitation within the claims, patentability cannot be found." (Office Action, p. 4, lines 3–5, 10–12, 17–21.) Each of these arguments is traversed.

With respect to (I), a prima facie case of obviousness exists when the claimed range overlaps or lies inside a range disclosed by the prior art. M.P.E.P. § 2144.05(I) (citing *In re Wertheim*, 541 F.2d 257 (CCPA 1976); *In re Woodruff*, 919 F.2d 1575 (Fed. Cir. 1990)). The prior art does not disclose elastic moduli or damping capacities within the claimed range, so there is no prima facie case of obviousness under 35 U.S.C. § 103(a).

In particular, the Office Action asserts that FIG. 19 of Sutton discloses a damping capacity (effective loss modulus) of 0.016, "which is which is greater than 0.015," and that 0.016 is therefore "greater than approximately 0.02 (insofar as 0.015 is approximately 0.02)." (Office Action, p. 3, lines 16–18; p. 14, lines 3–5.) With respect, this argument is disingenuous, and contradicts the plain, ordinary and customary meaning of Applicants' claims. M.P.E.P. § 2111.01 (words of a claim must be given their plain meaning unless inconsistent with the specification) (citing *In re Zletz*, 893 F.2d 319, 321 (Fed. Cir. 1989)). Applicants recite a damping capacity greater than approximately 0.02, and these words mean exactly what they say. *Id.* (ordinary, simple English words are construed to mean exactly what they say) (citing *Chef America, Inc. v. Lamb-Weston, Inc.*, 358 F.3d 1371, 1372 (Fed. Cir. 2004)). An effective damping capacity of 0.016, as disclosed by Sutton, does not read on a damping capacity greater than approximately 0.02, as claimed by Applicants, and claims 1, 16 and 19 are not obvious under 35 U.S.C. § 103(a).

Rejections based on obviousness, moreover, should rely on logic and sound scientific principles, and these principles do not extend to the theory that 0.016 is greater than approximately 0.02. See, e.g., M.P.E.P. § 2144.02 (in relying on theory, evidentiary support must be provided) (citing *In re Soli*, 317 F.2d 941 (CCPA 1963); *In re Grose*, 592 F.2d 1161, (CCPA 1979)). In addition, Applicants also specify that the recited damping capacities and moduli of elasticity are in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz, and at these frequencies Sutton's FIG. 19 discloses an effective damping capacity of less than 0.01, not 0.016.

More specifically, the recited frequency range corresponds to $\log_{10}$ frequency between about 3.78 and about 4.36 on Sutton's horizontal scale, where Sutton teaches effective tan(δ) values between 0.04 and 0.06. Sutton, FIG. 19. These correspond to effective damping capacities of

about 0.0064 to 0.0095,[2] and there is no reasonable interpretation of "greater than approximately 0.02," as claimed by Applicants, that can be read upon by an effective damping capacity of at most 0.0095, as taught by Sutton. See M.P.E.P. 2111.01 (claims to be given "broadest *reasonable* interpretation") (citing *In re Okuzawa*, 537 F.2d 545, 548 (CCPA 1976)).

In any case, as FIGS. 18 and 19 show that the data relied on by the Office Action were obtained by attaching a swatch of the damping material to a three-point test beam, in which case the dynamic storage modulus of the steel dominates and it is difficult to rigorously define tan(δ). Sutton, col. 23, lines 16–32 ("Because the dynamic storage modulus for steel largely dominates the effective E' for the system, although swatch properties may have a significant effect on this value, it is difficult to rigorously define an exact time-temperature superposition for tan δ."). Thus none of the effective tan(δ) data in FIGS. 18, 19, 22, 23, 26, 27, 31, 32, 35 and 36 are rigorously defined, and these data should not be relied on in making a rejection under 35 U.S.C. § 103(a).

When the damping materials are tested in the form of a fiber film, moreover, independently of the beam, as shown in FIGS. 20, 21, 24, 25, 28, 29, 33, 34, 37 and 38, the loss moduli are reduced by approximately two orders of magnitude.[3] The maximum disclosed elastic modulus for these data, regardless of frequency, is only about 0.5 GPa, twenty times less than the claimed minimum of 10 GPa. Sutton thus does not disclose materials "having damping capacities and moduli of elasticity within the claimed ranges," as asserted in the Office Action, and there is no prima facie case of obviousness under 35 U.S.C. § 103(a). (Office Action, p. 4, lines 10–13.)

With respect to (II), obviousness may also be found on the basis of optimization and routine experimentation when the general conditions of a claim are disclosed in the prior art. M.P.E.P. § 2144.05(II) (citing *In re Aller*, 220 F.2d 454, 456 (CCPA 1955)). Sutton, however, does not disclose the general conditions of Applicant's claims, because Sutton teaches that the effective tan(δ) data relied on by the Office Action are not rigorously defined, and that they do not represent real quantities for actual system components. (Sutton, col. 18, lines 25–34 ("For this reason, it

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[2] Damping capacity $\zeta = \delta / 2\pi$, so $\zeta = \tan^{-1}[\tan(\delta)]/2\pi$, where δ is scaled in radians. (See Application, ¶¶ 34–36.)
[3] For scale conversions, 1 GPa = 10^{10} dyne/cm^2.

desirable to define effective moduli for such a system, and an effective tan δ... *although not representative of real quantities for the components of the system*.") (emphasis added).

To the extent that Sutton's effective variables "allow the merits of different standard swatches to be determined," moreover, these are relative measures only, based on Sutton's particular three-point test beam geometry. *Id.* ("the response of such a system is a *relative* measure of damping material performance."); see also col. 13, lines 16–32 ("In order to provide appropriate *relative* comparisons, however, an effective tan δ is defined...") (emphasis added). Sutton's effective variables do not represent real quantities, and are not rigorously defined for the actual components of a suspension system, as claimed by Applicants. Sutton, col. 18, lines 16–24 ("without a complex analysis of strain in both the beam and swatch, *it is not possible to determine dynamic moduli for the two specific components of the system rigorously*.") (emphasis added). Sutton thus does not render claims 1, 16 and 19 obvious under 35 U.S.C. § 103(a), and these claims should be allowed.

With respect to (III), Applicants may rebut a prima facie case of obviousness based on overlapping ranges by providing evidence of criticality. M.P.E.P. § 2144.05(III) (citing *Woodruff*, 919 F.2d at 1575). Here, however, there is no prima facie case and the burden of proof does not shift to Applicants because (I) Sutton does not teach the recited ranges of damping capacity and elastic modulus, as described above; and (II) Sutton's effective variables do not represent real quantities, and are thus not applicable to the actual components of Applicants' invention as claimed.

Nonetheless, the record provides substantial evidence of criticality. In particular, the claimed damping capacities and elastic moduli are functions of vibrational frequency, and the vibrational frequencies are critical because they describe the critical frequencies of resonant modes of vibration. See, e.g., Arya, col. 1, lines 56–60 ("resonance at critical dynamic frequencies"); col. 2, lines 20–22 ("dynamic flutter and mode gains at critical frequencies"); see also Sutton, col. 1, lines 13–15; col. 12, line 15 – col. 13, line 21 (resonant vibration modes); Application ¶¶ 5, 14, 15, 23, 27, 34, 52, 53 (bending, torsion and sway modes).

While Sutton's effective variable analysis is restricted to a particular three-point test beam configuration, moreover, Sutton's FIGS. 20–29 and 31–38 also describe a sufficiently large number test results, undertaken both inside and outside the recited frequency range, to show that the

recited damping capacities, elastic moduli and mode frequencies are critical parameters. M.P.E.P. § 716.02(d)(II) (tests inside and outside the claimed range show criticality) (citing *In re Hill*, 284 F.2d 955 (CCPA 1960)). In addition, Applicants claim unexpectedly superior damping capacities, particularly when considering that the closest prior art data do not even represent real quantities for actual system components. M.P.E.P. § 716.02(b)(II) (superiority over closest prior art is evidence of nonobviousness) (citing *In re Chupp*, 816 F.2d 643, 646 (Fed. Cir. 1987)). With respect to the actual test swatch data, Applicants also show unexpected results because none of the test swatches has a loss modulus within a factor of twenty of the claimed range, as described above. M.P.E.P. § 716.02 (markedly superior results are unexpected) (citing *In re Wagner*, 371 F.2d 877, 884 (CCPA 1967); *Ex parte Gelles*, 22 USPQD 2d 1318, 1319 (Bd. Pat. App. & Inter. 1992)).

As taught by Arya, moreover, the issue of vibrational damping depends not only on material properties, but also on the free vibrating lengths between attachment points of the different suspension components, including hinge and gimbal components as recited in Applicants' claims. Arya, col. 1, line 65 – col. 2, line 28. In particular, attachment geometry contributes to dynamic flutter at the critical mode frequencies, and, as Applicants point out, this requires complex numerical techniques such as finite element analysis. Application, ¶¶ 48, 51–53.

The technique of finite element analysis is applied to notoriously unpredictable problems like weather prediction, and to complex elasticity and structural analysis problems including resonance mode vibrations in multi-component suspension assemblies, which are subject to numerical instabilities, complicated domain structures, and other unpredictable effects. See, e.g., WIKIPEDIA, Finite Element method (Appendix B). Even if Sutton's data represented real physical quantities, therefore (which they do not), there would still be no way to know how to apply these data to Applicants' claimed suspension assembly without performing Applicant's finite element analysis. This analysis, however, appears only in Applicants' disclosure, and not in the prior art, so the reconstruction of Applicants' claims is based on impermissible hindsight. M.P.E.P. § 2145(X)(A) (proper reconstruction takes into account only knowledge within the level of ordinary skill at the time the claimed invention was made and *does not include knowledge gleaned only from applicant's disclosure*) (citing *In re McLaughlin*, 443 F.2d 1392, 1395 (CCPA 1971)) (emphasis added).

Each of claims 1, 16 and 19 is nonobvious under 35 U.S.C. § 103(a), and should be allowed. Claims 2, 3, 5–12, 14, 15, 18, 20 and 26–32 are also allowable, as dependent from nonobvious base claims 1 and 16. M.P.E.P. § 2143.03 (in judging obviousness, all claim limitations must be considered; if an independent claim is nonobvious under § 103(a), any claim depending therefrom is nonobvious) (citing *In re Wilson*, 424 F.2d 1382, 1385 (CCPA 1970); *In re Fine*, 837 F.2d 1071 (Fed. Cir. 1988)).

b. Claims 1, 16 and 19 are nonobvious under 35 U.S.C. § 103(a) because the properties of the claimed structural damping materials would not have been predictable at the recited vibration mode frequencies.

Claim 1 and claim 16 each recite a suspension assembly having a beam component, a hinge component for connecting to an actuation arm and a gimbal component for carrying a transducer. Claim 1 specifies that the hinge component and the gimbal component each comprise a structural damping material having a modulus of elasticity greater than approximately 10 GPa and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 Hz and about 22,650 Hz. Claim 16 specifies that the hinge component consists essentially of a structural damping material having the recited damping capacity and modulus of elasticity. Claim 19, which depends from claim 16, specifies that the gimbal component comprises a structural damping material having the recited damping capacity and modulus of elasticity.

On pages 4 and 5, the Office Action asserts that it would have been obvious to arrive at the claimed structural damping materials in the course of routine experimentation and optimization, and, "absent any evidence of unexpected results within the frequency range recited by Applicant[s], the behavior of structural damping materials within the recited frequency range is presumed to have been predictable." (Office Action, p. 4, line 21 – p. 5, line 5.) It is an error, however, to conclude obviousness based upon a mere presumption of predictability, because the Office bears the burden of establishing a prima facie case. M.P.E.P. § 2144.08(II)(a) ("The PTO bears the burden of establishing a case of prima facie obviousness.") (citing *In re Bell*, 991 F.2d 781, 783 (Fed. Cir. 1993); *In re Rijckaert*, 9 F.3d 1531, 1532 (Fed. Cir. 1993); *In re Oetiker*, 977 F.2d 1443, 1445 (Fed. Cir. 1992); *Graham v. John Deere Co.*, 383 U.S. 1, 17 (1966)).

Because there is no prima facie case of obviousness, moreover, and as described above, there is no burden on Applicants to show unpredictability. *Id.* ("*If a prima facie case is established*, the burden shifts to applicant") (citing *Bell*, 991 F.2d at 783–84; *Rijckaert*, 9 F.3d at 1532; *Oetiker*, 977 F.2d at 1445) (emphasis added). Even where Applicants claim a particular range of mechanical properties, the Office still has the burden of making a prima facie case. See, e.g., M.P.E.P. § 2145(III) ("Applicants can rebut *a prima facie case of obviousness* based on overlapping ranges by showing [criticality]") (citing *Woodruff*, 919 F.2d at 1575) (emphasis added).

The Office Action also asserts that Applicants do not become inventors by following the teaching of Sutton "within a frequency band of concern for a different specific application," especially where there is market pressure toward minimization and predictability is presumed. (Office Action, p. 14, lines 12–15; p. 17, lines 5–6). With respect, Applicants do not follow Sutton's teaching and predictability may not be presumed. Applicants claim actual suspension components, not effective variables, as taught by Sutton, and Sutton's effective variables do not predictably teach to Applicant's actual suspension, as stated in the Sutton reference itself. In particular, while Sutton teaches that the effective loss modulus and effective tan(δ) "represent the values which would be obtained if a pure beam of material were tested which yielded exactly the same force response as that of the steel beam combined with the damping swatch," Sutton also explains that effective variables themselves are not physical, and thus do not represent "real quantities for the [actual] components of the system," as claimed by Applicants. Compare, e.g., Sutton, col. 18, lines 24–34 and 52–55.

More specifically, Sutton obtains the results shown in FIGS. 35 and 36 (and in FIGS. 18–29, 31–34, 37 and 38) by extrapolating data taken at low frequencies, no higher than 30 Hz, to frequencies beyond 22 kHz, more than three orders of magnitude higher. Sutton, col. 22, lines 2–10. The extrapolation, moreover, is based on a presumption that frequency scaling can be accurately modeled by varying the temperature, and then predicting the system's frequency-dependent damping properties based on temperature-dependent effects. Sutton, col. 21, line 63 – col. 23, line 53 (describing time-temperature superposition).

This method, however, is inherently unpredictable, and Sutton acknowledges this fact. See, e.g., Sutton, col. 18, lines 16–24 (not possible to rigorously determine dynamic moduli for

specific system components); col. 23, lines 26–30 ("it is difficult to rigorously define an exact time-temperature superposition for tan(δ)"). As a result, while materials with superior damping performance may yield higher effective loss moduli and tan(δ) by comparison, the effect is only relative to other samples on the same three-point test beam. Sutton, col. 18, lines 25–31, 56–67. Sutton's data do not represent real quantities for actual system components, as described above, and do not predictably model damping behavior for Applicants' suspension assembly as claimed.

In any case, Sutton limits analysis to the range of –10 to +40° C, over which the time-temperature superposition is generally accepted as applicable, and each of Sutton's results presumes a reference operating temperature of 20° C. Sutton, col. 7, line 1 – col. 9, line 42; col. 22, line 15 – col. 23, line 4; col. 25, lines 1–63; col. 28, line 56 – col. 29, line 53. The operating temperature of a typical hard disc drive, however, is 45–75° C, and Sutton's time-temperature superposition *presumes* that the damping properties depend upon temperature. See Sutton, col. 21, line 63 – col. 23, line 53; Application, ¶ 33. If Sutton's structural damping materials were applied to Applicants' suspension assembly, therefore, as suggested in the Office Action, they would *not* be expected to have the damping properties that were relied upon in rejecting Applicants' claims, and Sutton, in fact, provides no reliable means by which the actual properties could have been reliably predicted.

Claims 1, 16 and 19 are thus nonobvious under 35 U.S.C. § 103(a), and should be allowed. Claims 2, 3, 5–12, 14, 15, 18, 20 and 26–32 are also allowable, as dependent from nonobvious base claims 1 and 16. M.P.E.P. § 2143.03 (allowable under § 103 if dependent on nonobvious independent claim).

c. Claims 1 and 16 are nonobvious under § 103(a) because Arya teaches away from a separately formed hinge or gimbal, as recited in the claims, and away from attachment to the beam, as taught by Takagi.

Claim 1 and claim 16 each recite a suspension assembly having a beam component with a front end and a rear end, a hinge component for connecting to an actuation arm and a gimbal component for carrying a transducer. Claim 1 specifies that at least one of the hinge component and the gimbal component is separately made and attached to the beam component. Claim 16 specifies that the hinge component is separately made and attached to the rear end of the beam component.

In addressing the separately formed hinge or gimbal, as claimed by Applicants, the Office Action relies on Takagi for teaching "the many advantages of a separately formed construction." (Office Action, p. 17 lines 12–13.) The question, however, is not what Takagi suggests in isolation, but what Takagi's suggestion teaches to a person of ordinary skill who has read Arya, and who considers the prior at as a whole. M.P.E.P. § 2143.02(VI) (prior art must be considered as a whole, "including portions that would lead away from the claimed invention.") (citing *W.L. Gore & Assoc., Inc. v. Garlock, Inc.*, 721 F.2d 1540 (Fed. Cir. 1983)).

In particular, Arya teaches that the hinge and gimbal are not separately formed; instead, layers 36, 38 and 40 are formed continuously ("intact") through mount plate 102, hinge 108, beam 110 and gimbal 120. Arya, col. 6, ll. 33–35 ("third through fifth layers 36–40 intact, with the third layer 36 acting as a load bearing member"); see also col. 6, l. 41 – col. 7, l. 35; FIGS. 6–8. Third layer 36 thus forms an intact load-bearing structure all the way from load plate 102 through hinge 108 and load beam functional end 110 to gimbal system 120, including load/unload tang 140. Arya, col. 5, line 55 – col. 6; col. 6, lines 28–65; col. 7, lines 26–55. If the hinge or gimbal were separately formed, as suggested by the Office Action, and then attached to the beam, as claimed by Applicants, the load-bearing principles of Arya's entire suspension assembly would be destroyed. See, e.g., M.P.E.P. § 2143.01(VI) (proposed modification cannot change the principle of operation of prior art invention) (citing *In re Ratti*, 270 F.2d 810 (CCPA 1959)).

In addition, where Takagi teaches that spring member (hinge) 41 and flexure 50 are attached to load beam 31 by "laser welding or the like" (Takagi, ¶¶ 6, 44–46, 53, 57, 59), Arya teaches that the suspension assembly is weld free. Arya, col. 2, lines 26–28 (substantially if not completely weld free), lines 31–40 (weld free) col. 8, lines 1–2 (weld free); claims 1, 11, 21 (weld free). Thus Takagi cannot be combined with Arya, as suggested in the Office Action, and claims 1 and 16 are not obvious under 35 U.S.C. § 103(a). M.P.E.P. § 2145(X)(D)(2) (references *cannot be combined* where reference teaches away from their combination; it is *improper* to combine references where the references teach away from their combination) (citing *In re Grasselli*, 713 F.2d 731, 743 (Fed. Cir. 1983)) (emphasis added).

Arya also teaches a number of mechanical disadvantages, including multiple processing steps, additional processing cycles, fixturing, alignment tolerance, thermal distortion, flatness variation, reduced availability of real estate, and free vibrating lengths between the weld points. Arya, col. 1, line 65 – col. 2, line 28. While these teachings are directed to welding, moreover, they apply equally to other attachment methods, including adhesive attachments, because the intended purpose of the Arya invention is to provide a continuously formed suspension assembly that is not subject to free vibrating lengths between the attachment points, which contribute to dynamic flutter at critical mode frequencies. *Id.* See also M.P.E.P. § 2143.01(V) (proposed modification cannot render prior art invention unsatisfactory for its intended purpose) (citing *In re Gordon*, 733 F.2d 900 (Fed. Cir. 1984)).

On page 17, the Office Action asserts that Arya's teachings to the advantages of weld-free suspension are "far short of the sort of teaching away from separately formed suspension parts that would discourage a person of ordinary skill in the art from trying a not-completely-weld-free-suspension construction." (Office Action, p. 17, lines 8–12.) With respect, this is a mere conclusion, and does not support a rejection based on obviousness. M.P.E.P. § 2141(C)(III) ("Rejections on obviousness cannot be sustained by mere conclusory statements; instead, there must be some articulated reasoning with some rational underpinning to support the legal conclusion of obviousness.") (citing *KSR Int'l Co. v. Teleflex, Inc.*, 550 U.S. 398 (2007)).

In any case, the assertion is incorrect. In fact, Arya could hardly be more plain that a person of ordinary skill should not form a hinge or gimbal component separately, as suggested by Takagi, or attach such a separately formed component to a load beam, as claimed by Applicants, whether by welding or otherwise. While Takagi may suggest such a modification in the abstract, moreover, this is not the standard under 35 U.S.C. § 103(a). Under § 103(a), the prior art must be taken as a whole, and Takagi's suggestion must be viewed in light of Arya's teachings as well.

Arya has already fully considered the modification suggested by Takagi and proposed by the Office Action, and Arya has plainly rejected this modification in favor of a weld-free (attachment-free) continuous suspension assembly, in order to provide high performance without free vibrating lengths. See also, e.g., Arya, Title ("Weld Free High Performance Laminate Suspension").

Arya teaches away from a material aspect of Applicants' invention, and claims 1 and 16 are not obvious under 35 U.S.C. § 103(a). M.P.E.P. § 2144.05(III) (prima facie obviousness rebutted by showing that prior art teaches away from claimed invention in *any material respect*) (citing *In re Geisler*, 116 F.3d 1465, 1471 (Fed. Cir. 1997)) (emphasis added).

Claims 1 and 16 are nonobvious under 35 U.S.C. § 103(a), and should be allowed. Claims 2, 3, 5–12, 14, 15, 18–20 and 26–32 are also allowable, as dependent from nonobvious base claims 1 and 16. M.P.E.P. § 2143.03 (allowable under § 103 if dependent on nonobvious independent claim).

d. Claim 2 is not obvious under § 103(a) because the recited moduli of elasticity are not obvious over the prior art.

Claim 2 specifies that the structural damping materials recited in claim 1 have a modulus of elasticity greater than approximately 30 GPa in the claimed range of vibration mode frequencies, and claim 2 is nonobvious for at least the reasons given with respect to claim 1, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because the closest prior art fiber film samples have a maximum loss modulus of about 0.40 GPa, 75 times less than the recited value, and because the closest prior art effective damping capacities lie outside the claimed range. In any case, the closest prior art disclosures are non-physical and the behavior of the closest prior art materials cannot be predicted for Applicants' suspension assembly, as described above. See also, e.g., M.P.E.P. § 716.02 (marked improvement and markedly superior results are unexpected).

e. Claims 3 and 20 are not obvious under § 103(a) because the prior art teaches away from the same material for the hinge and the gimbal.

Claims 3 and 20 specify that the structural damping materials of the hinge and the gimbal components recited in claims 1 and 16 are substantially identical in composition, and these claims are nonobvious for at least the reasons given with respect to claims 1 and 16, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because Arya teaches that the hinge is formed from three layers and two of the layers are selectively removed to form the gimbal, so the materials are not substantially identical in composition. Arya, col. 6, lines 41–65 (hinge 108 with third through fifth layers 36, 38, 40 intact); col. 7, lines 36–55 (portions of fourth

and fifth layers 38, 40 selectively removed from gimbal region 120, to the right of line A–A in FIG. 6). See also M.P.E.P. § 2143.02(VI) (prior art leads away from invention).

f. Claim 6 is not obvious under § 103(a) because the prior art teaches away from a hinge made of a single material.

Claim 6 specifies that the entire hinge component is substantially made from the first structural damping material recited in claim 1, and claim 6 is nonobvious for at least the reasons given with respect to claim 1, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because Arya teaches that the hinge is formed from at least three different layers, not substantially from a first material. Arya, col. 6, lines 41–65 (hinge 108 formed by removing first and second layers 32, 34, leaving third through fifth layers 36, 38, 40 intact). M.P.E.P. § 2143.02(VI).

g. Claim 7 is not obvious under § 103(a) because the prior art teaches away from a gimbal made of a single material.

Claim 7 specifies that the entire gimbal component is substantially made from the second structural damping material recited in claim 1, and claim 7 is nonobvious for at least the reasons given with respect to claim 1, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because Arya teaches that the gimbal is formed from three layers, not substantially made from a second material. Arya, col. 7, lines 36–55 (portions of fourth and fifth layers 38, 40 selectively removed, but unselected portions remain). M.P.E.P. § 2143.02(VI).

h. Claim 8 is not obvious under § 103(a) because the prior art teaches attaching an external structural damping material.

Claim 8 specifies that the hinge component recited in claim 1 has no external structural damping material attached, and claim 8 is nonobvious for at least the reasons given with respect to claim 1, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because Sutton, which is relied on for structural damping materials, teaches attaching an external swatch of structural damping material. Sutton, col. 6, lines 36–44; FIG. 13 (rectangular swatch of damping material affixed to steel beam). M.P.E.P. § 2143.02(VI).

i. Claims 9 and 10 are not obvious under § 103(a) because the recited moduli of elasticity are not predicted by the prior art.

Claims 9 and 10 specify that the structural damping materials recited in claim 1 have moduli of elasticity greater than approximately 50 GPa in the claimed range of vibration mode frequencies, and these claims are nonobvious for at least the reasons given with respect to claim1, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because the closest prior art fiber film samples have a maximum loss modulus of about 0.40 GPa, 125 times less than the recited value, and because the closest prior art effective damping capacities lie outside the claimed range. In any case, the closest prior art disclosures are non-physical, and the behavior of the closest prior art materials cannot be predicted. M.P.E.P. § 716.02.

j. Claims 11, 18, 29 and 32 are not obvious under § 103(a) because the prior art teaches away from an alloy.

Claims 11, 18, 29 and 32 specify that the structural damping materials recited in claims 1 and 16 are alloys, and these claims are nonobvious for at least the reasons given with respect to claims 1 and 16, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because Sutton, which is relied on for structural damping materials, teaches that the structural damping materials are formed by incorporating a viscoelastic damping material into the matrix of a mechanically stable porous material, not that the structural damping materials are alloys. Sutton, col. 10, lines 10–15; FIGS. 1, 2. See also M.P.E.P. § 2143.02(VI).

k. Claim 14 is not obvious under § 103(a) because the prior art teaches away from attaching the hinge or gimbal through an adhesive.

Claim 14 specifies that at least one of the hinge and gimbal components recited in claim 1 is attached to the beam component through an adhesive, and claim 14 is nonobvious for at least the reasons given with respect to claim 1, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness Arya teaches an intact load-bearing structure, and because the disadvantages of welding, as taught by Arya, apply to adhesive attachments at well. Arya, col. 1, line 65 – col. 2, line 28. See also M.P.E.P. § 2143.02(VI).

l. Claim 15 is not obvious under § 103(a) because the prior art teaches away from attaching the hinge or gimbal by welding.

Claim 15 specifies that at least one of the hinge and gimbal components recited in claim 1 is attached to the beam component by welding, and claim 15 is nonobvious for at least the reasons given with respect to claim 1, above. M.P.E.P. § 2143.03. In addition, there is no prima facie case of obviousness because Arya teaches an intact load-bearing structure and because Arya teaches the disadvantages of welding. Arya, col. 1, line 65 – col. 2, line 28. See also M.P.E.P. § 2143.02(VI).

CONCLUSION

In view of the foregoing, it is respectfully requested that Applicants' Appeal from the final rejection of claims 1–3, 5–12, 14–16, 18–20 and 26–32 be granted. Each of claims 1–3, 5–12, 14–16, 18–20 and 26–32 is nonobvious under 35 U.S.C. § 103(a) and in condition for allowance, and a determination to that effect is in order.

Respectfully submitted,

KINNEY & LANGE, P.A.

Date: April 19, 2010

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NPL:els

Appendix A – Claims

1. (Previously presented) A head suspension assembly, comprising:

a beam component having a front end and a rear end;

a hinge component near the rear end of the beam component for connecting to an actuation arm; and

a gimbal component near the front end of the beam component for carrying a transducing head;

wherein the hinge component comprises a first structural damping material having a modulus of elasticity greater than approximately 10 gigapascals and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz and the gimbal component comprises a second structural damping material having a modulus of elasticity greater than approximately 10 gigapascals and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz; and

wherein at least one of the hinge component and the gimbal component is separately made and attached to the beam component.

2. (Previously presented) The head suspension assembly of claim 1, wherein the first structural damping material has a modulus of elasticity greater than approximately 30 gigapascals in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz, and the second structural damping material has a modulus of elasticity greater than approximately 30 gigapascals in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz.

3. (Previously presented) The head suspension assembly of claim 1, wherein the first structural damping material and the second structural damping material are substantially identical in composition.

4. (Canceled)

5. (Original) The head suspension assembly of claim 1, wherein the hinge component applies a preload on the transducing head through the beam component.

6. (Original) The head suspension assembly of claim 1, wherein the entire hinge component is substantially made from the first structural damping material only.

7. (Previously presented) The head suspension assembly of claim 1, wherein the entire gimbal component is substantially made from the second structural damping material only.

8. (Original) The head suspension assembly of claim 1, wherein the hinge component has no external structural damping material attached thereto.

9. (Previously presented) The head suspension assembly of claim 1, wherein the first structural damping material has a modulus of elasticity greater than approximately 50 gigapascals in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz.

10. (Previously presented) The head suspension assembly of claim 1, wherein the second structural damping material has a modulus of elasticity greater than approximately 50 gigapascals in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz.

11. (Previously presented) The head suspension assembly of claim 1, wherein the first structural damping material is an alloy.

12. (Previously presented) The head suspension assembly of claim 1, wherein the first structural damping material is a laminate comprising a stainless steel layer and a damping material layer.

13. (Canceled)

14. (Previously presented) The head suspension assembly of claim 1, wherein the at least one of the hinge component and the gimbal component is attached to the beam component through an adhesive.

15. (Previously presented) The head suspension assembly of claim 1, wherein the at least one of the hinge component and the gimbal component is attached to the beam component by welding.

16. (Previously presented) A head suspension assembly, comprising:

a beam component having a front end and a rear end;

a hinge component for connecting to an actuation arm, wherein the hinge component consists essentially of a first structural damping material having a modulus of elasticity greater than approximately 10 gigapascals and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz, and the hinge component is separately made and attached to the rear end of the beam component; and

a gimbal component near the front end of the beam component for connecting to a slider assembly carrying a transducer.

17. (Canceled)

18. (Original) The head suspension assembly of claim 16, wherein the first structural damping material is an alloy.

19. (Previously presented) The head suspension assembly of claim 16, wherein the gimbal component comprises a second structural damping material having a modulus of elasticity greater than approximately 10 gigapascals and a damping capacity greater than approximately 0.02 in a vibration mode having a frequency between about 6,010 hertz and about 22,650 hertz.

20. (Original) The head suspension assembly of claim 19, wherein the first structural damping material and the second structural damping material are substantially identical in composition.

21–25. (Canceled)

26. (Previously presented) The head suspension assembly of claim 1, wherein the first structural damping material is a composite.

27. (Previously presented) The head suspension assembly of claim 16, wherein the first structural damping material is a laminate comprising a stainless steel layer and a damping material layer.

28. (Previously presented) The head suspension assembly of claim 16, wherein the first structural damping material is a composite.

29. (Previously presented) The head suspension assembly of claim 19, wherein the second structural damping material is an alloy.

30. (Previously presented) The head suspension assembly of claim 19, wherein the second structural damping material is a laminate comprising a stainless steel layer and a damping material layer.

31. (Previously presented) The head suspension assembly of claim 19, wherein the second structural damping material is a composite.

32. (Previously presented) The head suspension assembly of claim 1, wherein the second structural damping material is an alloy.

Appendix B – Evidence

The following evidence was relied upon by the examiner as to grounds of rejection to be reviewed on appeal:

1. Arya et al., U.S. Patent No. 6,785,094, Application No. 10/131,553, filed April 24, 2002, issued August 31, 2004 (Arya).
2. Sutton et al., U.S. Patent No. 5,965,249, Application No. 08/908,619, filed August 7, 1997, issued October 12, 1999 (Sutton).
3. Takagi et al., U.S. Publication No. 2001/0008475, Application No. 09/793,410, filed February 26, 2001, abandoned January 23, 2005 (Takagi).

Arya and Sutton were entered into the record May 30, 2008 and Takagi was entered into the record September 3, 2009, each as cited by the examiner.

In addition, the following evidence is relied on by Applicants:

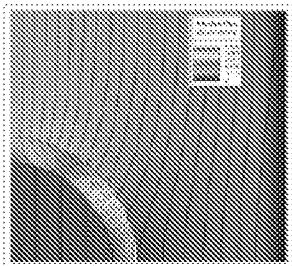
1. WIKIPEDIA, Finite Element method, http://en.wikipedia.org/wiki/Finite_element_method (December 3, 2009).

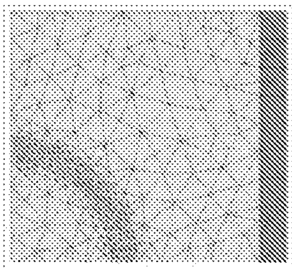
This evidence was entered into record with Applicants' Amendment filed December 3, 2009, as NPL, non-patent literature documents. A copy is provided.

Finite element method

From Wikipedia, the free encyclopedia

The **finite element method (FEM)** (sometimes referred to as **finite element analysis** (FEA)) is a numerical technique for finding approximate solutions of partial differential equations (PDE) as well as of integral equations. The solution approach is based either on eliminating the differential equation completely (steady state problems), or rendering the PDE into an approximating system of ordinary differential equations, which are then numerically integrated using standard techniques such as Euler's method, Runge-Kutta, etc.


2D FEM solution for a magnetostatic configuration (lines denote the direction of calculated flux density and colour - its magnitude)


2D mesh for the image above (mesh is denser around the object of interest)

In solving partial differential equations, the primary challenge is to create an equation that approximates the equation to be studied, but is numerically stable, meaning that errors in the input data and intermediate calculations do not accumulate and cause the resulting output to be meaningless. There are many ways of doing this, all with advantages and disadvantages. The Finite Element Method is a good choice for solving partial differential equations over complicated domains (like cars and oil pipelines), when the domain changes (as during a solid state reaction with a moving boundary), when the desired precision varies over the entire domain, or when the solution lacks smoothness. For instance, in a frontal crash simulation it is possible to increase prediction accuracy in "important" areas like the front of the car and reduce it in its rear (thus reducing cost of the simulation); Another example would be the simulation of the weather pattern on Earth, where it is more important to have accurate predictions over land than over the wide-open sea.

Contents

History

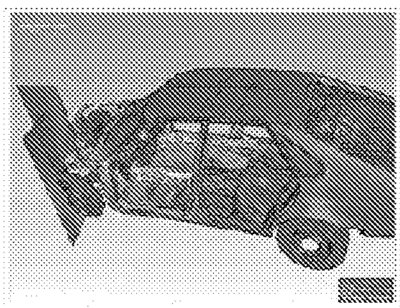
The finite-element method originated from the need for solving complex elasticity and structural analysis problems in civil and aeronautical engineering. Its development can be traced back to the work by Alexander Hrennikoff (1941) and Richard Courant (1942). While the approaches used by these pioneers are dramatically different, they share one essential characteristic: mesh discretization of a continuous domain into a set of discrete sub-domains, usually called elements.

Hrennikoff's work discretizes the domain by using a lattice analogy while Courant's approach divides the domain into finite triangular subregions for solution of second order elliptic partial differential equations (PDEs) that arise from the problem of torsion of a cylinder. Courant's contribution was evolutionary, drawing on a large body of earlier results for PDEs developed by Rayleigh, Ritz, and Galerkin.

Development of the finite element method began in earnest in the middle to late 1950s for airframe and structural analysis and gathered momentum at the University of Stuttgart through the work of John Argyris and at Berkeley through the work of Ray W. Clough in the 1960s for use in civil engineering. By late 1950s, the key concepts of stiffness matrix and element assembly existed essentially in the form used today. NASA issued request for proposals for the development of the finite element software NASTRAN in 1965. The method was provided with a rigorous mathematical foundation in 1973 with the publication of Strang and Fix's *An Analysis of The Finite Element Method*[1] has since been generalized into a branch of applied mathematics for numerical modeling of physical systems in a wide variety of engineering disciplines, e.g., electromagnetism and fluid dynamics.

Application

A variety of specializations under the umbrella of the mechanical engineering discipline (such as aeronautical, biomechanical, and automotive industries) commonly use integrated FEM in design and development of their products. Several modern FEM packages include specific components such as thermal, electromagnetic, fluid, and structural working environments. In a structural simulation, FEM helps tremendously in producing stiffness and strength visualizations and also in minimizing weight, materials, and costs.


Visualization of how a car deforms in an asymmetrical crash using finite element analysis.

FEM allows detailed visualization of where structures bend or twist, and indicates the distribution of stresses and displacements. FEM software provides a wide range of simulation options for controlling the complexity of both modeling and analysis of a system. Similarly, the desired level of accuracy required and associated computational

time requirements can be managed simultaneously to address most engineering applications. FEM allows entire designs to be constructed, refined, and optimized before the design is manufactured.

This powerful design tool has significantly improved both the standard of engineering designs and the methodology of the design process in many industrial applications.[2] The introduction of FEM has substantially decreased the time to take products from concept to the production line.[2] It is primarily through improved initial prototype designs using FEM that testing and development have been accelerated.[3] In summary, benefits of FEM include increased accuracy, enhanced design and better insight into critical design parameters, virtual prototyping, fewer hardware prototypes, a faster and less expensive design cycle, increased productivity, and increased revenue.[2]

Technical discussion

We will illustrate the finite element method using two sample problems from which the general method can be extrapolated. It is assumed that the reader is familiar with calculus and linear algebra.

P1 is a **one-dimensional** problem

$$\text{P1}\ :\begin{cases} u''(x)=f(x) \text{ in } (0,1), \\ u(0)=u(1)=0, \end{cases}$$

where f is given and u is an unknown function of x, and u'' is the second derivative of u with respect to x.

The **two-dimensional** sample problem is the Dirichlet problem

$$\text{P2}\ :\begin{cases} u_{xx}(x,y)+u_{yy}(x,y)=f(x,y) & \text{in } \Omega, \\ u=0 & \text{on } \partial\Omega, \end{cases}$$

where Ω is a connected open region in the (x,y) plane whose boundary $\partial\Omega$ is "nice" (e.g., a smooth manifold or a polygon), and u_{xx} and u_{yy} denote the second derivatives with respect to x and y, respectively.

The problem P1 can be solved "directly" by computing antiderivatives. However, this method of solving the boundary value problem works only when there is only one spatial dimension and does not generalize to higher-dimensional problems or to problems like $u + u'' = f$. For this reason, we will develop the finite element method for P1 and outline its generalization to P2.

Our explanation will proceed in two steps, which mirror two essential steps one must take to solve a boundary value problem (BVP) using the FEM.

- In the first step, one rephrases the original BVP in its weak, or variational form. Little to no computation is usually required for this step. The transformation is done by hand on paper.
- The second step is the discretization, where the weak form is discretized in a finite dimensional space.

After this second step, we have concrete formulae for a large but finite dimensional linear problem whose solution will approximately solve the original BVP. This finite dimensional problem is then implemented on a computer.

Variational formulation

The first step is to convert P1 and P2 into their variational equivalents, or Weak formulation. If u solves P1, then for any smooth function v that satisfies the displacement boundary conditions, i.e. $v = 0$ at $x = 0$ and $x = 1$, we have

(1) $\int_0^1 f(x)v(x)\,dx = \int_0^1 u''(x)v(x)\,dx.$

Conversely, if u with $u(0) = u(1) = 0$ satisfies (1) for every smooth function $v(x)$ then one may show that this u will solve P1. The proof is easier for twice continuously differentiable u (mean value theorem), but may be proved in a distributional sense as well.

By using integration by parts on the right-hand-side of (1), we obtain

(2)
$$\begin{aligned}\int_0^1 f(x)v(x)\,dx &= \int_0^1 u''(x)v(x)\,dx \\ &= u'(x)v(x)|_0^1 - \int_0^1 u'(x)v'(x)\,dx \\ &= -\int_0^1 u'(x)v'(x)\,dx = -\phi(u,v).\end{aligned}$$

where we have used the assumption that $v(0) = v(1) = 0$.

A proof outline of existence and uniqueness of the solution

We can loosely think of $H_0^1(0,1)$ to be the absolutely continuous functions of $(0,1)$ that are 0 at $x = 0$ and $x = 1$ (see Sobolev spaces). Such function are (weakly) "once differentiable" and it turns out that the symmetric bilinear map ϕ then defines an inner product which turns $H_0^1(0,1)$ into a Hilbert space (a detailed proof is nontrivial.) On the other hand, the left-hand-side $\int_0^1 f(x)v(x)dx$ is also an inner product, this time on the Lp space $L^2(0,1)$. An application of the Riesz representation theorem for Hilbert spaces shows that there is a unique u solving (2) and therefore P1. This solution is a-priori only a member of $H_0^1(0,1)$, but using elliptic regularity, will be smooth if f is.

The variational form of P2

If we integrate by parts using a form of Green's identities, we see that if u solves P2, then for any v,

$$\int_\Omega f v\, ds = -\int_\Omega \nabla u \cdot \nabla v\, ds = -\phi(u, v),$$

where ∇ denotes the gradient and $\cdot$ denotes the dot product in the two-dimensional plane. Once more ϕ can be turned into an inner product on a suitable space $H_0^1(\Omega)$ of "once differentiable" functions of Ω that are zero on $\partial\Omega$. We have also assumed that $v \in H_0^1(\Omega)$ (see Sobolev spaces). Existence and uniqueness of the solution can also be shown.

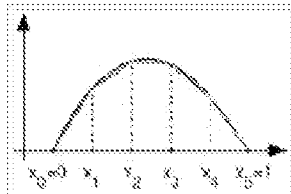
Discretization

The basic idea is to replace the infinite dimensional linear problem:

Find $u \in H_0^1$ such that
$\forall v \in H_0^1,\ -\phi(u, v) = \int fv$

with a finite dimensional version:

(3) Find $u \in V$ such that
$\forall v \in V,\ -\phi(u, v) = \int fv$



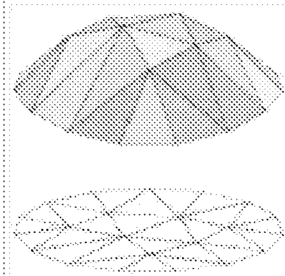
A function in H^1_0, with zero values at the endpoints (blue), and a piecewise linear approximation (red)

where V is a finite dimensional subspace of H_0^1. There are many possible choices for V (one possibility leads to the spectral method). However, for the finite element method we take V to be a space of piecewise linear functions.

For problem P1, we take the interval $(0, 1)$, choose n values of x with $0 = x_0 < x_1 < \ldots < x_n < x_{n+1} = 1$ and we define V by

$$V = \{v : [0, 1] \rightarrow \mathbb{R}\ : v \text{ is continuous, } v|_{[x_k, x_{k+1}]} \text{ is linear for } k = 0, \ldots, n, \text{ and } v(0) = v(1) = 0\}$$

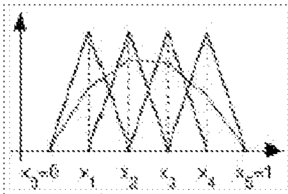
where we define $x_0 = 0$ and $x_{n+1} = 1$. Observe that functions in V are not differentiable according to the elementary definition of calculus. Indeed, if $v \in V$ then the derivative is typically not defined at any $x = x_k$, $k = 1, \ldots, n$. However, the derivative exists at every other value of x and one can use this derivative for the purpose of integration by parts.


A piecewise linear function in two dimensions.

For problem P2, we need V to be a set of functions of Ω. In the figure on the right, we have illustrated a triangulation of a 15 sided polygonal region Ω in the plane (below), and a piecewise linear function (above, in color) of this polygon which is linear on each triangle of the triangulation; the space V would consist of functions that are linear on each triangle of the chosen triangulation.

One often reads V_h instead of V in the literature. The reason is that one hopes that as the underlying triangular grid becomes finer and finer, the solution of the discrete problem (3) will in some sense converge to the solution of the original boundary value problem P2. The triangulation is then indexed by a real valued parameter $h > 0$ which one takes to be very small. This parameter will be related to the size of the largest or average triangle in the triangulation. As we refine the triangulation, the space of piecewise linear functions V must also change with h, hence the notation V_h. Since we do not perform such an analysis, we will not use this notation.

Choosing a basis



Basis functions v_k (blue) and a linear combination of them, which is piecewise linear (red).

To complete the discretization, we must select a basis of V. In the one-dimensional case, for each control point x_k we will choose the piecewise linear function v_k in V whose value is 1 at x_k and zero at every $x_j,\ j \neq k$, i.e.,

$$v_k(x) = \begin{cases} \frac{x - x_{k-1}}{x_k - x_{k-1}} & \text{if } x \in [x_{k-1}, x_k], \\ \frac{x_{k+1} - x}{x_{k+1} - x_k} & \text{if } x \in [x_k, x_{k+1}], \\ 0 & \text{otherwise}, \end{cases}$$

for $k = 1, \ldots, n$; this basis is a shifted and scaled tent function. For the two-dimensional case, we choose again one basis function v_k per vertex x_k of the triangulation of the planar region Ω. The function v_k is the unique function of V whose value is 1 at x_k and zero at every $x_j,\ j \neq k$.

Depending on the author, the word "element" in "finite element method" refers either to the triangles in the domain, the piecewise linear basis function, or both. So for instance, an author interested in curved domains might replace the triangles with curved primitives, and so might describe the elements as being curvilinear. On the other hand, some authors replace "piecewise linear" by "piecewise quadratic" or even "piecewise polynomial". The author might then say "higher order element" instead of "higher degree polynomial". Finite element method is not restricted to triangles (or tetrahedra in 3-d, or higher order simplexes in multidimensional spaces), but can be defined on quadrilateral subdomains (hexahedra, prisms, or pyramids in 3-d, and so on). Higher order shapes (curvilinear elements) can be defined with polynomial and even non-polynomial shapes (e.g. ellipse or circle).

Examples of methods that use higher degree piecewise polynomial basis functions are the hp-FEM and spectral FEM.

More advanced implementations (adaptive finite element methods) utilize a method to assess the quality of the results (based on error estimation theory) and modify the mesh during the solution aiming to achieve approximate solution within some bounds from the 'exact' solution of the continuum problem. Mesh adaptivity may utilize various techniques, the most popular are:

- moving nodes (r-adaptivity)
- refining (and unrefining) elements (h-adaptivity)
- changing order of base functions (p-adaptivity)
- combinations of the above (hp-adaptivity)

Small support of the basis

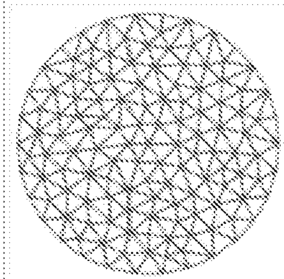
The primary advantage of this choice of basis is that the inner products

$$\langle v_j, v_k \rangle = \int_0^1 v_j v_k \, dx$$

and

$$\phi(v_j, v_k) = \int_0^1 v_j' v_k' \, dx$$

will be zero for almost all j,k. (The matrix containing $\langle v_j, v_k \rangle$ in the (j,k) location is known as the Gramian matrix.) In the one dimensional case, the support of v_k is the interval $[x_{k-1}, x_{k+1}]$. Hence, the integrands of $\langle v_j, v_k \rangle$ and $\phi(v_j, v_k)$ are identically zero whenever $|j-k| > 1$.



Solving the two-dimensional problem $u_{xx} + u_{yy} = -4$ in the disk centered at the origin and radius 1, with zero boundary conditions.
(a) The triangulation.

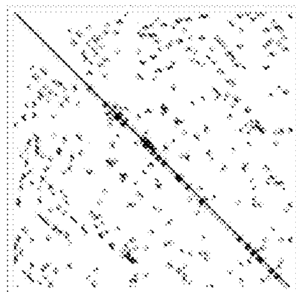
Similarly, in the planar case, if x_j and x_k do not share an edge of the triangulation, then the integrals

$$\int_\Omega v_j v_k \, ds$$

and

$$\int_\Omega \nabla v_j \cdot \nabla v_k \, ds$$

are both zero.

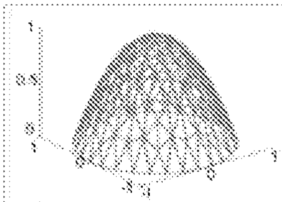


(b) The sparse matrix L of the discretized linear system.

Matrix form of the problem

If we write $u(x) = \sum_{k=1}^{n} u_k v_k(x)$ and $f(x) = \sum_{k=1}^{n} f_k v_k(x)$ then problem (3) becomes

(4) $-\sum_{k=1}^{n} u_k \phi(v_k, v_j) = \sum_{k=1}^{n} f_k \int v_k v_j$ for $j = 1, \ldots, n$.



(c) The computed solution, $u(x,y) = 1 - x^2 - y^2$.

If we denote by $\mathbf{u}$ and $\mathbf{f}$ the column vectors $(u_1, \ldots, u_n)^t$ and $(f_1, \ldots, f_n)^t$, and if we let $L = (L_{ij})$ and $M = (M_{ij})$ be matrices whose entries are $L_{ij} = \phi(v_i, v_j)$ and $M_{ij} = \int v_i v_j$ then we may rephrase (4) as

(5) $-L\mathbf{u} = M\mathbf{f}$.

As we have discussed before, most of the entries of L and M are zero because the basis functions v_k have small

support. So we now have to solve a linear system in the unknown $\mathbf{u}$ where most of the entries of the matrix L, which we need to invert, are zero.

Such matrices are known as sparse matrices, and there are efficient solvers for such problems (much more efficient than actually inverting the matrix.) In addition, L is symmetric and positive definite, so a technique such as the conjugate gradient method is favored. For problems that are not too large, sparse LU decompositions and Cholesky decompositions still work well. For instance, Matlab's backslash operator (which uses sparse LU, sparse Cholesky, and other factorization methods) can be sufficient for meshes with a hundred thousand vertices.

The matrix L is usually referred to as the stiffness matrix, while the matrix M is dubbed the mass matrix.

General form of the finite element method

In general, the finite element method is characterized by the following process.

- One chooses a grid for Ω. In the preceding treatment, the grid consisted of triangles, but one can also use squares or curvilinear polygons.
- Then, one chooses basis functions. In our discussion, we used piecewise linear basis functions, but it is also common to use piecewise polynomial basis functions.

A separate consideration is the smoothness of the basis functions. For second order elliptic boundary value problems, piecewise polynomial basis function that are merely continuous suffice (i.e., the derivatives are discontinuous.) For higher order partial differential equations, one must use smoother basis functions. For instance, for a fourth order problem such as $u_{xxxx} + u_{yyyy} = f$, one may use piecewise quadratic basis functions that are C^1.

Another consideration is the relation of the finite dimensional space V to its infinite dimensional counterpart, in the examples above H_0^1. A conforming element method is one in which the space V is a subspace of the element space for the continuous problem. The example above is such a method. If this condition is not satisfied, we obtain a nonconforming element method, an example of which is the space of piecewise linear functions over the mesh which are continuous at each edge midpoint. Since these functions are in general discontinuous along the edges, this finite dimensional space is not a subspace of the original H_0^1.

Typically, one has an algorithm for taking a given mesh and subdividing it. If the main method for increasing precision is to subdivide the mesh, one has an h-method (h is customarily the diameter of the largest element in the mesh.) In this manner, if one shows that the error with a grid h is bounded above by Ch^p, for some $C < \infty$ and $p > 0$, then one has an order p method. Under certain hypotheses (for instance, if the domain is convex), a piecewise polynomial of order d method will have an error of order $p = d + 1$.

If instead of making h smaller, one increases the degree of the polynomials used in the basis function, one has a p-method. If one combines these two refinement types, one obtains an hp-method (hp-FEM). In the hp-FEM, the polynomial degrees can vary from element to element. High order methods with large uniform p are called spectral finite element methods (SFEM). These are not to be confused with spectral methods.

For vector partial differential equations, the basis functions may take values in $\mathbb{R}^n$.

Comparison to the finite difference method

The finite difference method (FDM) is an alternative way of approximating solutions of PDEs. The differences between FEM and FDM are:

- The most attractive feature of the FEM is its ability to handle complicated geometries (and boundaries) with relative ease. While FDM in its basic form is restricted to handle rectangular shapes and simple alterations thereof, the handling of geometries in FEM is theoretically straightforward.
- The most attractive feature of finite differences is that it can be very easy to implement.
- There are several ways one could consider the FDM a special case of the FEM approach. One might choose basis functions as either piecewise constant functions or Dirac delta functions. In both approaches, the approximations are defined on the entire domain, but need not be continuous. Alternatively, one might define the function on a discrete domain, with the result that the continuous differential operator no longer makes sense, however this approach is not FEM.
- There are reasons to consider the mathematical foundation of the finite element approximation more sound, for instance, because the quality of the approximation between grid points is poor in FDM.
- The quality of a FEM approximation is often higher than in the corresponding FDM approach, but this is extremely problem dependent and several examples to the contrary can be provided.

Generally, FEM is the method of choice in all types of analysis in structural mechanics (i.e. solving for deformation and stresses in solid bodies or dynamics of structures) while computational fluid dynamics (CFD) tends to use FDM or other methods like finite volume method (FVM). CFD problems usually require discretization of the problem into a large number of cells/gridpoints (millions and more), therefore cost of the solution favors simpler, lower order approximation within each cell. This is especially true for 'external flow' problems, like air flow around the car or airplane, or weather simulation in a large area.

Various types of finite element methods

Generalized finite element method

The Generalized Finite Element Method (FEM) uses local spaces consisting of functions, not necessarily polynomials, that reflect the available information on the unknown solution and thus ensure good local approximation. Then a partition of unity is used to "bond" these spaces together to form the approximating subspace. The effectiveness of GFEM has been shown when applied to problems with domains having complicated boundaries, problems with micro-scales, and problems with boundary layers.[4]

hp-FEM

The hp-FEM combines adaptively elements with variable size *h* and polynomial degree *p* in order to achieve exceptionally fast, exponential convergence rates[5].

XFEM

Main article: Extended finite element method

Spectral methods

Main article: Spectral method

Meshfree methods

Main article: Meshfree methods

Discontinuous Galerkin methods

Main article: Discontinuous Galerkin method

See also

- Direct stiffness method
- Boundary element method
- Discrete element method
- Finite element machine
- Finite element method in structural mechanics
- Galerkin method
- Multiphysics
- Patch test
- Rayleigh-Ritz method
- List of finite element software packages
- Multidisciplinary design optimization

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5. ^ P. Solin, K. Segeth, I. Dolezel: Higher-Order Finite Element Methods, Chapman & Hall/CRC Press, 2003

External links

- Multiphysics Glossary (http://www.comsol.com/products/multiphysics/glossary/) (Glossary of Multiphysics and Finite Element Modeling terms by COMSOL)
- NAFEMS (http://www.nafems.org) -- The International Association for the Engineering Analysis Community
- IFER (http://homepage.usask.ca/~ijm451/finite/fe_resources/fe_resources.html) -- Internet Finite Element Resources - an annotated list of FEA links and programs

- Workshop "The Finite Element Method in Biomedical Engineering, Biomechanics and Related Fields" (http://www.uni-ulm.de/uni/intgruppen/fem/)
- Finite Element Analysis Resources (http://www.feadomain.com) - Finite Element news, articles and tips
- COMSOL Multiphysics Finite Element Analysis Software (http://www.comsol.com/) - Official site
- CAD, Finite Element Analysis (Abaqus,Ansys), CAE, Programming (http://www.caejournal.com) - FEM, CAD, Programming, discussion forums
- Finite Element Books (http://www.solid.ikp.liu.se/fe/index.html) - books bibliography
- Mathematics of the Finite Element Method (http://math.nist.gov/mcsd/savg/tutorial/ansys/FEM/)
- Finite Element Methods for Partial Differential Equations (http://web.comlab.ox.ac.uk/oucl/work/endre.suli/fem.ps) - Lecture notes by Endre Süli
- FEM AVI-gallery at CompMechLab site, St.Petersburg State Polytechnical University, Russia (http://www.eng.fea.ru/ANSYS_LSDYNA_AviGallery.html)
- Intro to FEA (http://knol.google.com/k/ife-deutschland/fea/1ycxqvmgvg8x/4#)
- Introduction to FEA for EM modeling (http://www.cvel.clemson.edu/modeling/tutorials/techniques/fem/finite_element_method.html) (includes list of currently available software)
- Finite Element modeling of light propagation (http://www.elis.ugent.be/ELISgroups/lcd/research/bpm.php)
- World Association of Fatigue, Durability and Fracture Mechanics - Fatigue for Finite Element Models (http://www.fatiguedurability.com)

Retrieved from "http://en.wikipedia.org/wiki/Finite_element_method"

Categories: Continuum mechanics | Finite element method | Numerical differential equations | Partial differential equations | Structural analysis

Appendix C – Related Proceedings

–None–

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4. *In re Geisler*, 116 F.3d 1465 (Fed. Cir. 1997
5. *In re Rijckaert*, 9 F.3d 1531 (Fed. Cir. 1993).
6. *In re Bell*, 991 F.2d 781 (Fed. Cir. 1993).
7. *In re Oetiker*, 977 F.2d 1443 (Fed. Cir. 1992)
8. *In re Woodruff*, 919 F.2d 1575 (Fed. Cir. 1990).
9. *In re Zletz*, 893 F.2d 319 (Fed. Cir. 1989).
10. *In re Fine*, 837 F.2d 1071 (Fed. Cir. 1988).
11. *In re Gordon*, 733 F.2d 900 (Fed. Cir. 1984).
12. *In re Grasselli*, 713 F.2d 731 (Fed. Cir. 1983).
13. *W.L. Gore & Assoc., Inc. v. Garlock, Inc.*, 721 F.2d 1540 (Fed. Cir. 1983).

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14. *In re Grose*, 592 F.2d 1161 (CCPA 1979).
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